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PROPELLANT VALVES OF LIQUID-PROPELLANT ROCKET ENGINES

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Foreign Technology Division
Wright-Patterson Air Force Base, Ohio

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PROPELLANT VALVES OF LIQUID-PROPELLANT
ROCKET ENGINES

by

A. I. Edel'man

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13. ABSTRACT

The book deals with problems in design and finish-off of automatic fuel units of liquid propellant rocket engines, specifications for these units and methods for their achievement. Analyzed are the effects of various factors on performance and reliability of units, as well as a series of standard designs of fuel valves. Given are testing methods, calculation principles and operating regulation of units. The book was written for specialists working in the field of design and operation of liquid-propellant rocket engines and their units; it can be useful also to college professors and students.

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Combustion Chamber						
Turbopump Feed System						
Oxidizer Tank						
Gas Generator						
Valve Disk						
Pneumatic Valves						
Stainless Steel Kh18NIOT						
Soft Aluminum						
Bellows						

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U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
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Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

* ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

DESIGNATIONS

$F_{э}$	= F_{e}	= $F_{effective}$
$D_{ср}$	= D_{av}	= $D_{average}$
$D_{вн}$	= D_{in}	= $D_{internal\ (or\ inner)}$
$D_{н}$	= D_{ex}	= $D_{external\ (or\ outer)}$
кГ	= kgf	= kilograms force
$D_{с}$	= D_{s}	= D_{seat}
$D_{пр}$	= D_{fp}	= $D_{flow\ passage}$
$w_{ст}$	= w_{w}	= w_{wall}
$k_{д}$	= k_{d}	= $k_{dynamic}$
$\Delta p_{зам}$	= Δp_{meas}	= $\Delta p_{measured}$
$Q_{ном}$	= Q_{nom}	= $Q_{nominal}$
$\Delta p_{прив}$	= Δp_{red}	= $\Delta p_{reduced}$
$p_{2\,доп}$	= $p_{2\,add}$	= $p_{2\,additional}$
$\gamma_{т}$	= γ_{prop}	= $\gamma_{propellant}$
$\gamma_{в}$	= γ_{wat}	= γ_{water}
$P_{упр}$	= P_{cont}	= $P_{control}$
$P_{вх}$	= P_{in}	= P_{inlet}
$P_{вых}$	= P_{out}	= P_{outlet}
ЭПВ	= EPV	= electropneumatic valve
$P_{н}$	= P_{boost}	= P_{boost}
$R_{см}$	= $R_{смесь}$	= $R_{mixture}$

Designations (continued)

$R_{ост}$	= $R_{оставшееся}$	= $R_{remaining}$
сек	= sec	= seconds
ат	= at	= technical atmosphere
мин	= min	= minutes
час	= h	= hour
кГ/см	= kgf/cm	= kilogram force/centimeter
мм	= mm	= millimeter

PROPELLANT VALVES OF LIQUID-PROPELLANT
ROCKET ENGINES

A. I. Edel'man

Propellant valves of liquid propellant rocket engines. A. I. Edel'man, Moscow, "Mashinostroyeniye," 1970.

This book discusses problems of the design and final adjustment of structures of automatic propellant assemblies of liquid propellant rocket engines (LPRE's), the requirements shown for such assemblies and methods for providing for these needs.

The influence of various factors on the efficiency and reliability of the assemblies is investigated. A number of typical constructions of propellant valves are examined. Test methods, the fundamentals of design and rules for operating the assemblies are given.

The book is intended for specialists, working in the area of the design and operation of liquid-propellant rocket engines and their assemblies; it may also be useful to instructors and students of schools of higher education.

13 Tables, 140 illustrations, 21 bibliographical entries.

Reviewer, Doctor of Technical Sciences, Professor G. S. Skubachevskiy. Editor, Engineer K. Ya. Zaytsewa.

FORWARD

The achievements of Soviet science and technology in the area of the mastery of space have found worldwide acknowledgement. The successes have been achieved by Soviet scientists, engineers and workers to a significant degree due to the creation of highly reliable liquid propellant rocket engines (LPRE's) [ЖРД].

The efficiency of LPRE is in many respects determined by the proper functioning of its individual systems, units and assemblies.

One of the most important and complex problems, arising in the creation of a liquid propellant rocket engine is the assurance of a satisfactory course of the transient processes.

The transient processes to a large extent determine the degree of reliability of the engine. It is sufficient to say that the majority of defects and breakdowns which can arise during final adjustment of the engine, occur precisely during unstable modes - when starting (mainly) or shutting down the engine. The character of a transient process and the law of change in time of the fuel supply conditions are largely determined by the operation of the control system and the engine regulation assemblies: the starting and shut-off propellant valves, the electropneumatic valves, pressure-release valves and other assemblies.

The sequence of operation of the propellant valves, the speed of their opening and closing, the nature of the change in the hydraulic resistance and, to a certain degree, the sealing of individual units - all this enters into the number of factors, determining the character of the transient process, the character of the change of conditions in the combustion chamber. Consequently, the further perfection and development of LPREs are closely connected with the need for deep analysis and comprehensive research of the assemblies of the control system, the reliable and stable operation of which is one of the decisive factors of the reliability of the engine on the whole.

In this work we have discussed fundamental problems, arising during the creation and final adjustment of structures of propellant valves of liquid-propellant rocket engines.

These problems are elucidated from the viewpoint of providing for the requirements imposed on the assemblies, the fulfillment of which is essential for the creation of an efficient engine.

The book consists of eight chapters.

Chapter 1 deals with the classification of propellant valves, the area of their application, justification of the approach to the selection of a type of valve, the basic requirements for assemblies, and the mutual influence of one assembly on another.

Chapter 2 describes certain characteristic structures of assemblies of repeated (pneumatic valves) and one-time (pyrotechnic valves) action, as well as their peculiarities, which determine the efficiency of the valves.

In Chapters 3, 4 and 5 an examination is made of the operation of movable and stationary seals, and analysis is made

of their pros and cons, as regards high aggressiveness of the fuels, low temperatures and other factors.

Chapters 6 and 7 give basic calculations, made in the process of designing propellant valves; describe the principal layouts of test benches, on which investigative final adjustments of the assemblies are carried out; give the procedure for carrying out tests; and also discuss problems of the technology of safety during operation of test benches.

The last chapter is devoted to ensuring the stability of operation of pneumatic valves of LPREs using high- and low-boiling propellants.

The pages of this book reflect the results of the joint work of many years of a group of persons, carried out with the participation of the now-deceased N. P. Alekhin and N. S. Lyakhonskiy, who made a great creative contribution to the theory and practice of the design of LPRE valves.

The author expresses his thanks to Professor G. S. Skubachevskiy, who made a number of valuable observations in reviewing the book, and also to Professor N. B. Rutovskiy for his useful advice in surveying the manuscript.

The author will be greatful to all readers, who inform the publishing house of the book's shortcomings noted by them.

GENERAL INFORMATION ON PROPELLANT VALVES

1.1. CLASSIFICATION OF PROPELLANT VALVES

Propellant valves are those assemblies, designed to control the flow rate of propellant components through an engine's propellant lines. Using propellant valves the flow passage cross-sectional area of the corresponding main lines is partially or completely opened (or closed).

Propellant valves must eliminate the possibility of supplying fuel and oxidizer to the combustion chamber and to the LPRE gas generator before starting and after engine shut off, as well as provide the possibility of transition from one mode to another.

The speed of opening (closing) of a propellant valve of a LPRE and its construction in great measure determine the character of the occurrence of the transient processes during the period of starting or final cut off of the engine.

Propellant valves are installed on the main lines, supplying propellant to the combustion chamber[1], on the plumbing, feeding

[1]The valves installed behind the pump, through which the basic propellant flow enters the combustion chamber, are called the main propellant valves.

the gas or steam generator, and on the lines which supply propellant to the pressurized tank system. Depending on the valve's use, there are various sizes and constructions.

In the sample schematic of a rocket engine using a turbopump feed system (Fig. 1.1) there are shown main fuel valve 23 and main oxidizer valve 24, placed between combustion chamber 26 and fuel pump 21 and oxidizer pump 19. At the inlet to the turbopump assembly are shown diaphragm valves 15 and 16, preventing the feeding of fuel to the pumps before the engine starts operating. The working medium for turbine 20 is supplied through valve 18 from gas generator 14, the propellant for which passes from tanks 10 and 11 through valves 13. Pressurization of tanks 10 and 11 of the gas generator is accomplished through reducer 6 of the propellant feed system into the gas generator (affecting the number of revolutions of the turbopump assembly); to eliminate the possibility of mixing of the propellant component vapors of the gas generator (after operation) there are check valves 7, while drainage valves 9 during this period connect propellant tanks 10 and 11 with the atmosphere, preventing the formation of excess pressure of the component vapors; during engine operation they are closed. Valves 12 are used for filling tanks 10 and 11 with propellant.

Valve 22 and check valve 25 are designed to scavenge the main propellant lines of the combustion chamber by an inert gas or air with the purpose of eliminating traces of propellant components.

Diaphragm valves 8 are positioned at the inlet and at the outlet of the gas generator's propellant tanks.

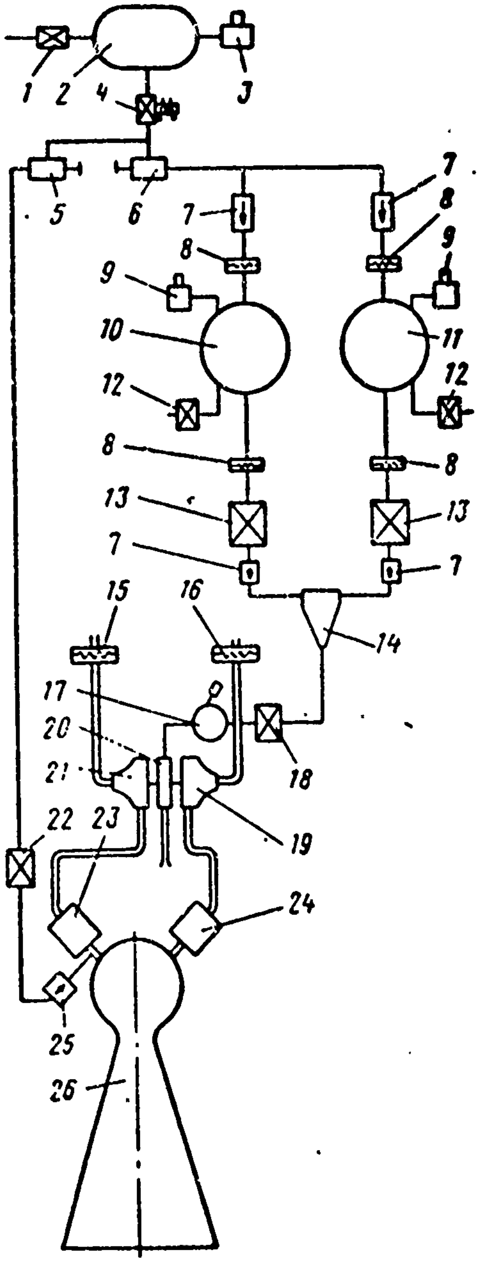


Fig. 1.1. Diagram of a LPRE with turbopump feed system and autonomous gas generation circuit of a two-component gas generator: 1 - compressed-air accumulator filler valve; 2 - compressed air cylinder; 3 - drainage valve; 4 - electropneumatic valve; 5 - main-line purging reducer; 6 - reducer for systems supplying propellant to gas generator; 7 - check valves; 8 - diaphragm valves; 9 - drainage valves; 10 - gas generator oxidizer tank; 11 - gas generator fuel tank; 12 - filler valves; 13 - gas generator propellant valves; 14 - gas generator; 15 - fuel diaphragm valve; 16 - oxidizer diaphragm valve; 17 - thrust regulator; 18 - gas generator gas valve; 19 - oxidizer pump; 20 - turbine; 21 - fuel pump; 22 - scavenging valve; 23 - main fuel valve; 24 - main oxidizer valve; 25 - scavenging check valve; 26 - combustion chamber.

As can be seen from the diagram, the engine has a great number of propellant valves for various purposes (the diagram does not show valves for filling, pressurization and drainage of the basic propellant tanks of the vehicle). Propellant valves of various engines may have completely different construction and operating principles. Thus, main valves 23 and 24, designed for one type

of engine, must both open and closed, passing propellant into the combustion chamber and cutting off the flow; main valves designed for other types of engines only have to close; before starting they are open, while the starting of the engine is provided for by the opening of valves 15 and 16. In this case valves 23 and 24 are called cut off valves, while valves 15 and 16 are called starting valves. Propellant valves in some engine types must ensure several operations, but in others only a single triggering is sufficient.

A system of classifying propellant valves is presented in Fig. 1.2.

Depending on the number of operations propellant valves of LPRE are divided into two basic types: repeating valves and one-time valves.

Engines on space vehicles, engines installed on aircraft as an accelerator or as the basic power unit, must ensure repeated starting, i.e., the valves of such engines must be of the repeating type.

Other types of LPRE must ensure one-time starting. These are engines of various types of rockets, engines of the primary stages of the launch vehicles of spacecraft and others. The valves of these engines may be of the one-time type.

Although only a single triggering is, in substance, required of each valve of such a LPRE, repeating valves are also widely used for these engines; during the assembly and in the preparation for starting the engine the functioning of the valves is checked. In developing a new engine prototype such valves permit repeated starting of the engine without removing it from the test bench facility.

A repeating valve accomplishes both opening and closing of the propellant main line, i.e., serves simultaneously as a starting valve and as a cut off valve.

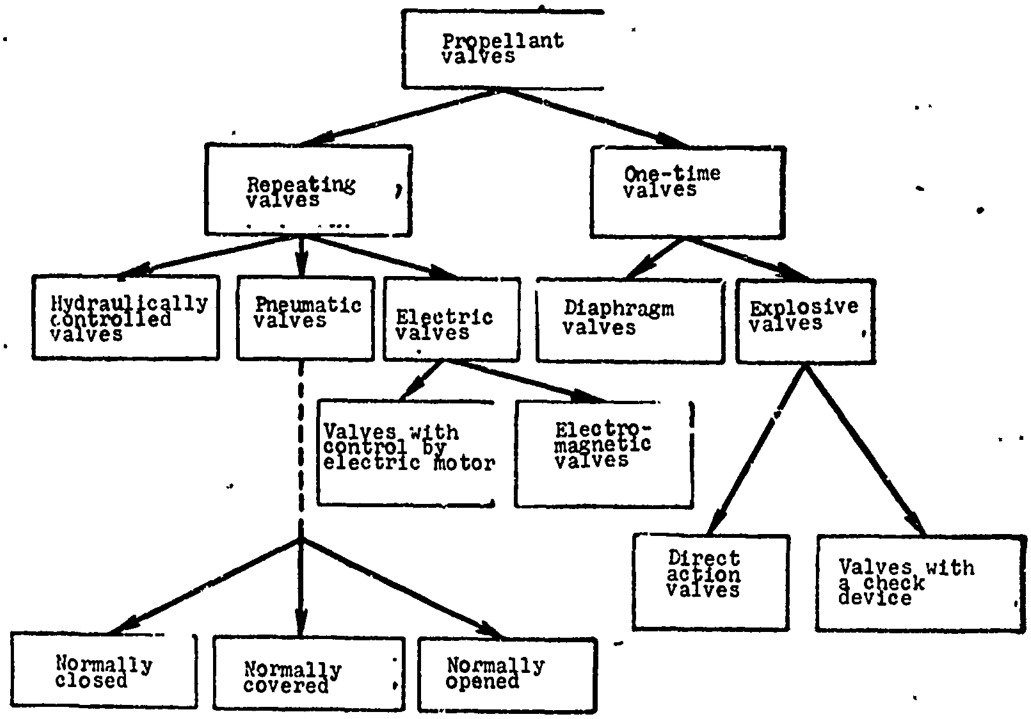


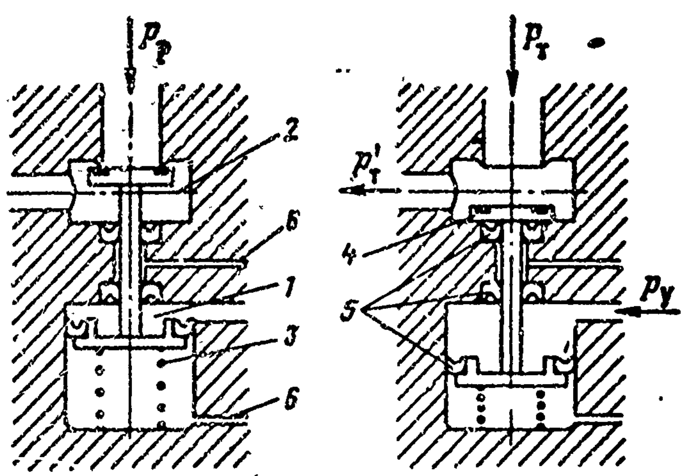
Fig. 1.2. Diagram of the classification of propellant valves of LPREs.

One-time valves are used only once, to accomplish a single operation: either to open, or to close a main line. In order to carry out both of these operations, it is necessary to have two valves or a block of one-time valves. Checking the operation of one-time valves before starting the engine is impossible.

Figure 1.1 does not show how energy is transmitted to one valve or another. Different types of valves are controlled in different ways: pneumatic valves - by the energy of a compressed gas; electromagnetic valves and valves controlled by an electric motor - by electricity; hydraulic valves and diaphragm valves -

by the pressure of the propellant itself; explosive valves - by the pressure of solid-reactant gases, which form during explosion of the blast cartridges. However, in this or that engine the majority of propellant valves are controlled identically - for example, one engine will use mainly pneumatic valves, another - explosive valves.

The most popular repeating valves are pneumatic valves - assemblies controlled by a compressed gas, which is supplied with the aid of an electropneumatic valve (EPV). With the supplying of the gas pressure into the controlling cavity the lock mechanism of the propellant valve moves, closing (or opening) the flow passage cross-sectional area in the main propellant line.



a) (a) б) (b)

Fig. 1.3. Diagram of a normally closed valve: a - no controlling air - valve is closed; b - controlling air is supplied - valve is opened; 1 - controlling cavity; 2 - liquid cavity; 3 - spring; 4 - valve disk; 5 - seal rings; 6 - drainage openings; p_T propellant pressure at the valve inlet (during flow); p'_T pressure at the valve outlet; p_p calculated static propellant pressure at the inlet; p_y normal gas pressure in the controlled cavity.

Pneumatically controlled propellant valves consist of the following basic elements (Figs. 1.3, 1.4 and 1.5): the assembly housing, in which units and parts are mounted; a locking mechanism, or simply a valve, capable of separating the outlet cavity from the inlet; the separating elements (sealed rings, rings or bellows), separating the controlled cavity from the propellant cavity; a return spring, employed for the return travel of the locking mechanism after the removal of the control pressure.

Pneumatically controlled repeating valves, in construction principally similar to propellant valves of LPREs, are employed as bench equipment on stands designed for testing the engine as a whole, and on devices designed for the final adjustment of individual units of the engine. These test bench valves are designed to supply (remove) a propellant (water, compressed air) to the tested unit. Valves designed for stopping the flow (the cut off of a component) are called cut off valves. Sometimes both of these functions are fulfilled by the same bench valve.

Pneumatic valves of an engine, as a rule, have only one controlling pressure, the action of which causes the movement of the locking mechanism (for opening or for closing; return travel is under the effect of the spring).

Test bench pneumatic valves, which have a very long service life, sometimes have two controlling pressures (two controlling cavities), providing movement of the locking mechanism both for closing and for opening. Such valves are called dual-control pneumatic valves (see Fig. 2.12).

In low-thrust engines electric valves (EV), controlled by electromagnets, for example, the valve described further in Fig. 2.14, are employed as repeating propellant valves.

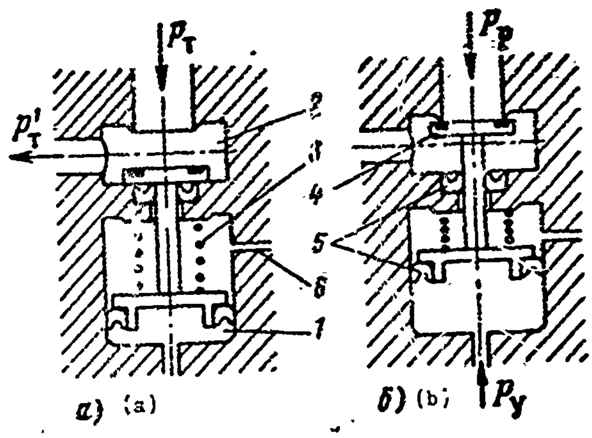


Fig. 1.4. Diagram of a normally opened valve: a - no controlling air - valve is open; b - controlling air is supplied - valve is closed. Designations are the same as in Fig. 1.3.

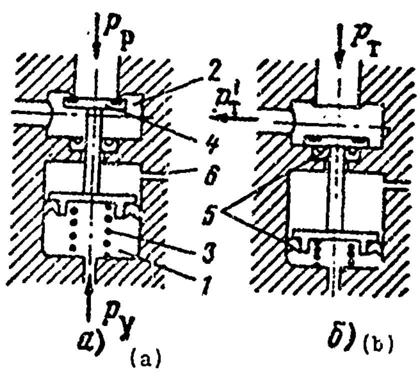


Fig. 1.5. Diagram of a normally covered valve: a - propellant pressure is low: valve is closed even in the absence of controlling air; controlling air is supplied - valve is closed even with increased propellant pressure; b - controlling air is discharged; valve is opened by the propellant pressure. Designations are the same as in Fig. 1.3.

The existing types of pneumatically controlled propellant valves, depending on the initial position of the locking mechanism, are divided into normally closed, normally opened and normally covered valves.

Normally closed pneumatically controlled propellant valves are those assemblies which in the absence of a compressed gas in the controlling cavity are closed by the force of a spring (see Fig. 1.3). Such valves must be sealed with a certain

(calculated) propellant pressure at the inlet p_p. When gas with pressure p_y is supplied to the controlling cavity the valves are opened.

Normally open pneumatically controlled valves are those assemblies, which in the absence of a compressed gas in the controlling cavity are opened by the force of a spring (see diagram shown in Fig. 1.4). When there is a certain gas pressure in the controlling cavity the valves are closed; the nominal pressure of the compressed gas p_y must ensure sealing of the valve at a given inlet static propellant pressure p_p.

Normally covered valves are those which in the absence of a compressed gas in the controlling cavity and in the absence of pressure in the liquid cavity are closed by the effect of a spring force. When compressed gas is supplied to the controlling cavity the force which effects the closing is increased (see diagram shown in Fig. 1.5).

The nominal pressure of the compressed gas p_y must ensure the sealing of the valve at a given inlet propellant pressure p_p. When there is no pressure in the controlling cavity, sealing is ensured only at a propellant pressure significantly less than p_p. Such valves are opened with an increase in propellant pressure and with the absence of the pressure in the controlling cavity.

Thus, the difference in the operation of normally opened and normally covered valves consists in the different directions of action of the force of the spring; the difference in the operation of normally closed and normally covered valves consists in the different directions of action of the force of the compressed gas.

One-time valves of any construction contain an explosive element (a tapered rod pin, a cut bead, a diaphragm), the

destruction of which proceeds under the action of the pressure of a fluid or of a solid-reactant gases. This leads to the opening or closing of the flow passage cross-sectional area of the main line.

LPRE employ one time assemblies, in the majority of cases controlled explosive cartridges, i.e., explosive valves.

During the burning of the pyrotechnic composition of the explosive cartridge combustion products under high pressure are formed. This pressure causes movement of the locking mechanism and leads to the opening or closing of the assembly. For each operation (opening, closing) there must be a separate assembly.

When an engine uses an automatic explosive device as the starting elements, diaphragms which explode either under the action of the propellant pressure or with the aid of explosive cartridges are frequently employed.

Propellant explosive valves are more frequently used as cut off devices for shutting off the engine, i.e., as normally opened assemblies.

Explosive propellant valves can be divided into two groups;
a) direct-action explosive valves;
b) explosive valves with a check device.

Direct action explosive valves are those assemblies, in which the pressure of the gases formed as a result of combustion of the pyrotechnic composition destroys the burst element, connected with the locking mechanism, and moves the latter; sometimes a spring and a pressure drop in the liquid (propellant) contributes to this movement.

The basic elements of a direct-action explosive valve are: the jacket, the explosive cartridges, the movement system with the locking device, the burst element, usually made as a whole with a part of the movement system.

The diagram of a normally open direct-action explosive valve is shown in Fig. 1.6. During the operation of explosive cartridge 1 the gas pressure on piston 2 will shear rim 3 of valve 4 and the valve, in moving, becomes wedged in seat 5, locking the path of the liquid.

Direct action propellant explosive valves are ordinarily of small size and are used to close main lines with an internal diameter of up to 20-30 mm. Explosive valves of this type are used not only as cut off valves (i.e., normally open), but also sometimes as starting valves (i.e., normally closed).

Explosive valves with check devices (or with check explosive valves) are those assemblies, in which during the detonation of the explosive cartridge by the pressure of the solid-reactant gases, the special lock pin is pulled out, holding the locking mechanism of the valve; under the action of the spring and the flow of liquid the locking mechanism moves to the stop.

Basic elements of a check explosive valve are: the jacket of the assembly, the explosive cartridge, the movement system with the locking mechanism, the spring, the unit of the check device with the shear element (here the shear element is made in the lock pin).

A typical schematic of an explosive valve with a lock pin is shown in Fig. 1.7.

Explosive valves with check devices are used as cut off valves (normally open) for placement on main lines with large diameters[1].

The locking mechanism in explosive check valves is always shifted in the direction of movement of the propellant.

Depending on where the solid-reactant gases go after triggering, the explosive valves may be divided into valves with isolation of the solid-reacting gases and valves with drainage of the solid-reactant gases.

In assemblies with isolation of the gases after triggering of the explosive cartridge the combustion products remain in a closed volume (see below Fig. 2.20, area A); the solid-reacting gases do not go outside, into the space surrounding the explosive valve. In explosive valves with drainage of the gases the triggering of the cartridge is accompanied by the bursting of the flame through the drainage openings (for example, through connector fitting 5, in Fig. 2.19). Explosive valves with drainage, as a rule, are structurally simpler and easier to adjust.

Sometimes individual explosive valves, each of which has its own special functions, are combined into a single block of explosive valves. For example, the triggering of one of the explosive cartridges provides for the passage of a small quantity of propellant, the triggering of another explosive cartridge provides for the passage of the total expenditure of the propellant, while the triggering of the third leads to the stopping of the flow.

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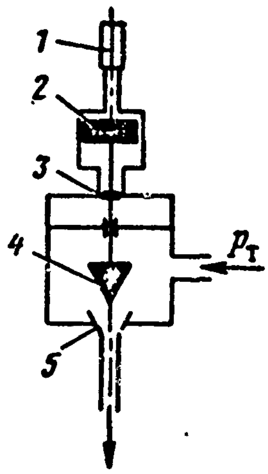
[1]In order to use an explosive check valve as a normally closed valve, it must be equipped with a diaphragm, which is burst under the pressure of the propellant after the operation of the check device, which holds the stop of the diaphragm.



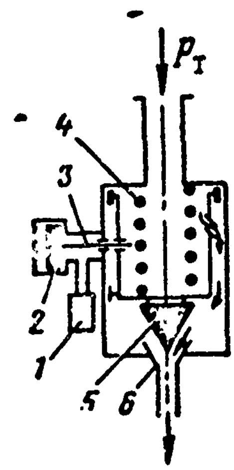


Fig. 1.6. Diagram of a normally open direct-action explosive valve: 1 - explosive cartridge; 2 - piston; 3 - shearing rim; 4 - valve; 5 - sear.

Fig. 1.7. Diagram of a normally open explosive valve with a check device: 1 - explosive cartridge; 2 - piston; 3 - lock pin; 4 - spring; 5 - valve; 6 - seat.

The direction of movement of the locking mechanism in the closing of the valve relative to the flow direction of the liquid has a significant effect on the conditions of the application and on the construction of valves.

Depending on the direction of movement of the locking mechanism direct action valves and return valves are distinguished.

Direction action valves are those assemblies, in which during closing the flow direction of the liquid (the propellant) is opposite to the direction of movement of the movable system (see diagrams shown in Fig. 1.3, 1.4 and 1.5)[1]. Such valves are frequently called "valves which close against the flow".

[1]During opening the flow of the propellant aids the opening.

Return valves are those assemblies, in which the direction of the flow of liquid during closing coincides with the direction of movement of the moving system (Fig. 1.8). The speed of movement of the locking mechanism of such a valve is, as a rule, close to the speed of the flow of propellant inside the assembly; closing of the valves is ordinarily accompanied by phenomenon of hydraulic impact in the supply line. The opening of pneumatic valves of similar type where there is the pressure of the propellant at the inlet to the valve sometimes leads to the breakage of the valve stem due to the great inertial forces at the moment of termination of movement of the moving system. Return valves may include explosive valves, the schematics of which are shown in Fig. 1.6 and 1.7. Such valves are sometimes called "Valves which close with the flow".

Sometimes propellant valves depending on the operating scheme are divided into valves with relief of the closing force (valves of the balanced type) and valves without relief (valves of the unbalanced type).

Valves of the unbalanced type are those assemblies, in which the driving force must exceed the force from the maximum propellant pressure, acting on the manifold cross sectional area (see schematics shown in Figures 1.3, 1.4, 1.6).

Valves of the balanced type are those assemblies, in which the driving force may be less than the force from the maximum propellant pressure, acting on the inlet manifold cross-sectional area (Fig. 1.9).

Sometimes in place of the terms "balanced" and "unbalanced" valves the terms "relief-type valves" and "nonrelief-type valves" are used.

Figure 1.9 shows one of the possible schemes of a pneumatic valve of the balanced type. The propellant passes inside the valve to large openings e in the valve disk. The force from the propellant pressure, absorbed by the controlling air during the closing of the assembly, is considerably less than the force required with an ordinary valve system with the same intake area, for example, for a valve manufactured according to the schematic shown in Fig. 1.5. Actually, with a nonbalanced valve the force R from the propellant pressure p_T during closing would amount to

$$R = p_T F_{BX} - (p_T - \Delta p)(F_{BX} - f_{Ш}) = p_T f_{Ш} + \Delta p (F_{BX} - f_{Ш}),$$

where F_{BX} is the area of the upper seat;

$F_{Ш}$ is the area of the rod;

Δp is the pressure differential between the valve inlet and outlet.

This force must be less than the force M, arising from the controlling pressure p_y:

$$M = p_y f_M > p_T f_{Ш} + \Delta p (F_{BX} - f_{Ш}),$$

where f_M is the area of the seal ring of the controlling cavity (according to the outer diameter).

With the valve schematic shown in Fig. 1.9 we have

$$R_1 = p_T F_{BX} + (p_T - \Delta p)(F_2 - F_{BX}) - \\ - (p_T - \Delta p_1)(F_2 - f_{Ш}) = \Delta p (F_{BX} - F_2) + \Delta p_1 (F_2 - f_{Ш}) + p_T f_{Ш},$$

where F_2 is the area of the second (large) seat;

Δp_1 is the pressure drop at openings e.

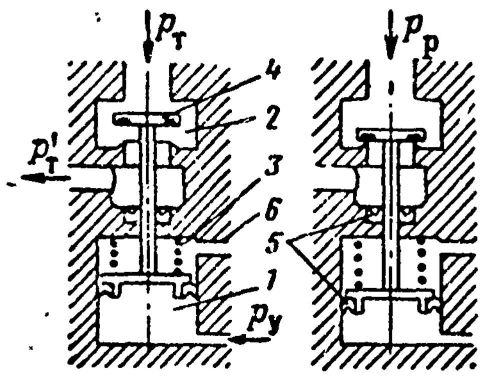


Fig. 1.8. Schematic of a return valve (closing with the flow) a - valve is open; b - valve is closed. Designations are the same as in Fig. 1.3.

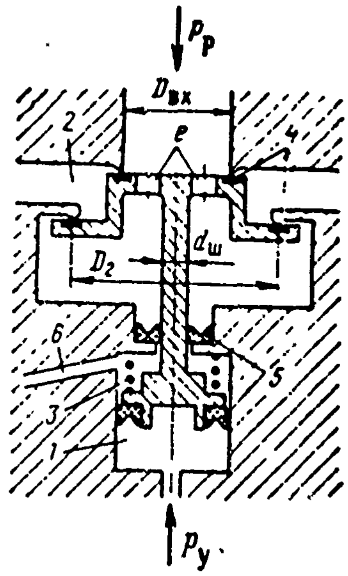


Fig. 1.9. Diagram of a relief-(balanced-) type valve. Designations are the same as in Fig. 1.3.

When $\Delta p_1 = 0$, as ordinarily occurs,

$$R_1 = p_т f_ш - \Delta p(F_2 - F_{вх}).$$

The force R_1 can be substantially less than R and M. The assembly shown below in Fig. 2.8 is constructed according to a similar system.

Valves of the balanced type are used only when the drive force (the spring, the pressure of decompressed gas) during closing must overcome the considerable force from the propellant, which occurs in direct action valves. Assemblies of the balanced type have until now been encountered only in pneumatic valves, but such systems are also possible for direct-action explosive valves.

The use of unloading allows us to reduce the gas pressure in the controlling cavity or to reduce its area, and also to use a weaker spring. This permits us to reduce the dimensions and the weight of the assembly.

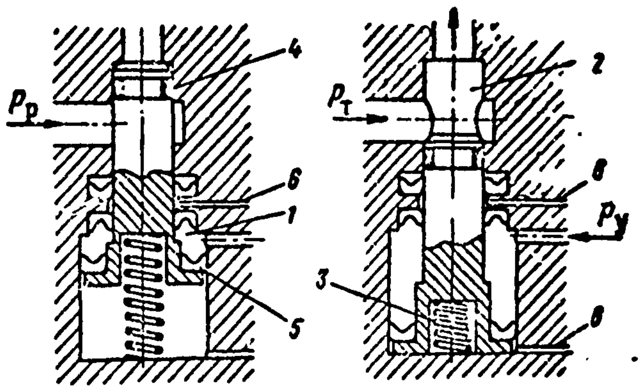


Fig. 1.10. Diagram of a normally closed valve with sealing of the locking mechanism by annular rings
a - valve is closed (controlling pressure is released);
b - valve is open (controlling pressure is supplied);
1 - controlling cavity; 2 - liquid cavity; 3 - spring;
4 - sealing ring; 5 - seal; 6 - drainage holes.

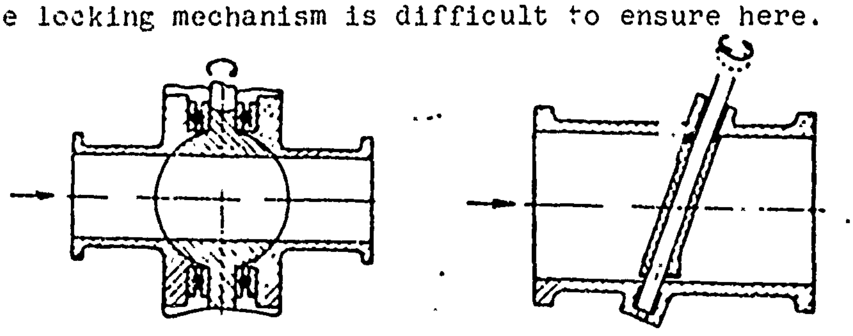
All the above cited pneumatic assemblies, as well as the constructions described in Section 2.1, are disk type valves, where the locking mechanism is a forward moving disk, usually with a seal made from a soft material, where the inlet and outlet tubes of the valve are situated at an angle (as a rule, the angle is 90°).

Pneumatic valves, where the sealing of the locking mechanism is ensured not by the seating of a disk on a flat seat, but by the seating of a glass along a cylindrical surface with the aid of rubber rings of circular cross section 4 (Fig. 1.10) show great promise (especially for normally closed assembly). Similar seals by rubber rings are widely used in assemblies of aircraft engines.

The advantage of such a seal in normally closed valves over ordinary seals of disk type is the fact that in ordinary designs during the prolonged action of a load (from the force of the springs) the material of the seal of the disk may leak, and be deformed, which will result in the loss of airtightness, while with the manufacture of the construction according to the scheme shown in Fig. 1.10 these factors are significantly less.

In principle it is possible to manufacture assemblies of repeating action of other types, where the locking mechanism executes not reciprocating, but rotational movement - the rotation around this or another axis. Such assemblies may improve, valves with a spherical or conical rotating plug where the rotation of the plug opens or closes the openings in the plug for the passage of propellant (valves of similar type are in successful operation; see, for example, Fig. 1.11a; throttle-type valves with a shutter, where the rotation of a shutter relative to the axis passing through its diameter varies the flow through cross section of the assembly - this construction is similar to the butterfly valve in the carburetor of an internal combustion engine (see Fig. 1.11b).

As a result of the fact that the inlet to the assembly and the outlet from it are placed on the same axis, these valves possess minimum hydraulic resistance; however, complete sealing of the locking mechanism is difficult to ensure here.



а) (a) б) (b)

Fig. 1.11. Diagrams of propellant valves: a - with spherical plug; b - throttle-type.

The construction of the drive of such assemblies (they are not necessarily pneumatically controlled)[1] is significantly more complex than the construction of the drive of disk type valves: there is a great quantity of parts which move relative to one another; the number of seals is increased; there is friction in the locking mechanism itself. All of this leads unavoidably to a reduction in the reliability of the assembly.

There are disk type valves, with inlet and outlet tubes situated on one axis - so-called uniflow valves[2]; they possess all the advantages of disk type valves and have low hydraulic resistance besides. These valves have comparatively long length and because of this they are not suitable for putting in one-chamber engines.

Apart from the given classification, propellant valves can be divided into fuel valves and oxidizer valves. The type of propellant, its specific characteristics have a significant effect not only on the materials used for the manufacture for the parts, but also on the entire system and construction of the assembly. An especially important effect is exerted by the type of propellant on the construction of repeating valves. The construction of explosive valves is less affected by the type of propellant.

1.2. THE CRITERIA DETERMINING THE SELECTION OF THE TYPE OF PROPELLANT VALVE

The selection of the engine control system (pyrotechnic or pneumatic) is determined primarily by the engine design, by the type of vehicle into which a given engine is placed, and depends on whether single or repeated triggering of the valves is required. Moreover, one should be guided by the following considerations.

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[1]An electric motor drive is possible.

[2]Uniflow valves of the single-action type and explosive valves of the check type are in widespread use.

The use of explosive starting elements (pyrotechnic diaphragms, normally closed explosive valves) ensures a high degree of sealing and therefore permits a very long storage time for the flight vehicle witn the installed engine and with the filled components.

With a control system using an automatic pyrotechnics system it is not necessary (indeed, it is impossible) to control the operation of the assemblies after their installation on the engine. Only periodic monitoring of the continuity of the electrical circuits is possible and feasible. Preparation for the starting of such engines requires very little time.

The layout of an engine with an automatic pyrotechnics system is ordinarily simpler than that of an engine with an automatic pneumatic system, and the number of assemblies in it is less (there are no controlling gas - reducer system, electropneumatic valve, vents, or check valves).

The triggering time for an explosive valve (after the moment of giving it the command to burst the explosive cartridge) is considerably less than the time for triggering a pneumatic valve. As a result of the speed of their action, the divergence in the triggering times of various samples of explosive valves of identical construction (especially of direct-action explosive valves) is very small with respect to absolute value. This favors the stability of the transient processes of the engine.

Another advantage of explosive valves is the relative simplicity of their construction, the usually somewhat smaller size and lower weight in comparison with pneumatically controlled valves.

Explosive valves, as a rule, do not require the use of such rubber seals, which determine the reliability of the construction,

and also of special consistent lubricants, and the assurance of the working reliability of the assembly within a wide range of temperatures is simplified, i.e., precisely those problems are eliminated the solution of which creates the greatest difficulties in the construction and manufacture of reliable pneumatically controlled valves.

A disadvantage of explosive valves is the fact that the possibility of controlling the transient processes is lessened with their use. With the utilization of a pneumatic system it is not difficult to change the time characteristics of the processess of supplying propellant to the combustion chamber by means of changing the speed of opening of the propellant pneumatic valves and providing a smoother or a more abrupt pressure increment. When using a pyrotechnic automatic system it is possible to change only the moments of opening (or closing) of one valve relative to another. In this way, the layout of a LPRE with pneumatic propellant valves is more flexible in operation.

The basic disadvantage of explosive assemblies is the impossibility of checking the triggering of each specimen of the assembly, as it is done in pneumatically controlled valves. The operational reliability of a whole batch of pyrotechnic assemblies must be judged only from the results of tests of several "control" specimens[1], randomly selected from the batch. This makes it necessary to have a great quantity of similar "control" assemblies and requires the especially careful finishing of the construction of the valve.

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[1]They can not be used after testing.

As a result of the high closing speed of explosive valves hydraulic shocks may arise in the system. These hydraulic shocks reach a significant value (see below) and may bring about the destruction of the conduits. Therefore, when using a pyrotechnic system it is often necessary to take special measures to avoid hydraulic shocks (the creation of special hydraulic brakes inside the assembly, the installation of dampers, the reduction in mode before closing the explosive valves and so forth).

When using explosive valves with drainage of the solid-reactant gases it is necessary to keep the following in mind: since the jet of the flame coming out of the drainage openings is composed of products of incomplete combustion, it cannot be assumed that oxidizer vapors were in the vicinity of the explosive valves, because this can lead to conflagration. In view of this, it is more desirable to use explosive valves with isolation of the solid-reactant gases. However, in similar constructions it is usually difficult to exclude the possibility of penetration of the solid-reactant gases into the liquid cavity of the assembly. Yet in a number of cases such penetration of gases presents no danger - for example, for fuel valves, for valves which handle nitric acid, in those instances where the solid-reactant gases immediately fall into the large volume of the propellant.

The use of two-position or multiposition explosive valves has a number of advantages over the application of some simple explosive valves. The combining of individual assemblies into a single block, with one jacket, as a rule, reduces weight (in comparison with the total weight of simple explosive valves), reduces the overall number of joints (seals), operating under high pressure, and reduces the overall dimensions. However, the manufacture of multiposition explosive valves is technologically more complex.

Multiposition explosive valves are employed in relatively small flow passage cross-sectional areas of propellant main lines.

A pyrotechnic system cannot be used as bench equipment, since in this case after each test it is necessary to replace the pyrotechnic assemblies.

It should be noted that the final adjustment of a construction of a pyrotechnic valve (dispite its simplicity) requires a large number of samples of assemblies (i.e., high material expenditures), since after each triggering of the explosive valve the greater part of its components are put out of commission; but when using a pneumatic assembly on one sample of it, it is possible to make many triggerings and checks, by simulating various operational conditions.

Moreover, it should be kept in mind that the application of a pneumatic automatic system allows a reduction in the period of adjustment of the engine on the whole, because of the possibility of conducting repeated fire tests without removing the engine from the stand.

Any restart of the engine when using a pyrotechnics system is impossible without removing the engine from the stand, because after an unsuccessful test start (even if only one of the explosive cartridges worked) the engine cannot be used again until the replacement of the assembly, and this requires the removal of the propellant from the engine cavity, flushing down, drying and so forth. Even if the defect which caused a failure during starting is insignificant and easily eliminated, the removal of the engine from the stand to replace assemblies and further operations require a lot of time.

A setup using a pneumatic system is more flexible and permits repeated starting without the replacement of the assemblies after the elimination of the effect.

For the engines of space vehicles, where repeated starting is required, a pyrotechnic automatic system is not suitable in the majority of cases, since it leads to considerable complication on the system and to a large increase in weight. However, in this case a very high degree of sealing of the locking mechanisms is required of the pneumatic assemblies: during prolonged intervals between engine starts the propellant must not seep through the seals of the assemblies.

When pneumatic valves are employed the selection of this or that type of assembly is made on the following considerations.

From the condition of safety of operation on the ground (before starting) it is preferable to use normally closed valves. With normally open valves in the case of a drop in the gas pressure in the controlling cavity (as a result of damage of the conduits which supplied the controlling gas, or the failure of some reducer or electropneumatic valve, EPV on the path of the gas and so forth) the spontaneous opening of the valves will occur, which can lead to serious consequences.

However, the construction of normally closed disk-type valves, operating at high propellant pressures, is usually complicated. Prolonged (during storage) existance of the locking mechanism of the valve in the closed position under high spring pressures (especially in the case of direct action valves) can lead to the destruction of the working capability of the seals - and to the loss in hermeticity. When using normally closed valves, which close with the flow, i.e., without strong springs, hydraulic shocks are to be feared. Therefore in repeat-action LPRE normally open or normally covered valves of the direct-action type are most often used.

Normally closed disk-type return valves should be used with operation at low propellant velocities, when hydraulic shock phenomena present no danger.

The use of normally closed single-action valves for LPRE permit a reduction in the gas supply in the pressure containers onboard the flight vehicle, since the consumption of gas takes place only during launch, when its supply may be provided from ground-based containers.

Balanced-type valves are employed for the abrupt cessation of the propellant flow, moving under high pressure.

For bench valves (from considerations of safety) the use of normally closed valves is recommended. Often employed in bench facilities are two tandem valves, one of which is normally closed, and the other - normally opened. Their alternating opening during starting and alternating closing during the cessation of operation provides reliability, safety of operation in hermeticity; however, the hydraulic resistance of the communications and the cost of the equipment are increased.

The type of propellant has a significant effect on the solution of the problem of selecting a specific valve design. The propellant properties determine the layout of the valve, the material of the parts, and the form of lubrication. Thus, the selection of lubrication for low-boiling propellants is very difficult. At low temperature the lubricant "freezes" and loses its antifriction properties; in a medium of liquid, and especially gaseous, oxygen the majority of lubricants are inclined to spontaneous combustion and explosion. At the same time the application of a lubricant is essential to carry out normal assemblage, and in the case of using pneumatic assemblies lubrication is also desirable to ensure the operation of the valve during checks before starting the engine (before oxygen touches the valve).

1.3. TECHNICAL REQUIREMENTS FOR PROPELLANT VALVES AND METHODS FOR ENSURING THEM

The requirements which are imposed on propellant valves may be arbitrarily divided into two groups:

a) requirements for the assembly itself - autonomous requirements;

b) "design requirements" - i.e., requirements caused by the interaction of assemblies during operation of the engine.

By the term "design requirements" we mean the specific requirements for the operation of the assemblies on a <u>given type of engine</u>, which determine the duration and character of the processes of opening and closing of the propellant pneumatic valves during their mutual operation, as well as the correspondence of the flow rates per second of both components of the propellant between themselves during transient conditions, i.e., the provision of those necessary conditions, which strictly determine the operating regime of an engine unit of a given type in the transient processes.

The "design requirements" for any propellant valve must not be examined apart from the engine in which it is set up, apart from the conditions of its starting, stopping and all possible operating conditions. Here it is essential to carefully take into consideration the sequence and the peculiarities of operation of all the assemblies of the engine.

The checking for the satisfaction of the "design requirements" and in the majority of cases even the very posing of the question of the need for these or other requirements becomes possible only during final adjustments of the entire engine as a whole.

By the term "autonomous requirements" we mean the requirements imposed on a given assembly, taken separately, which are not connected with its use on this or another type of engine.

The basic, most general, autonomous requirements for all the assemblies of liquid-propellant rocket engines are the requirements for reliability and low weight.

Operational reliability for an assembly of a LPRE is the main, decisive criterion, which subordinates to itself all the other factors which characterize the design; the weight, manufacturability, hydraulic resistance, cost and so forth.

The operating reliability of an assembly is primarily determined by its fundamental layout, by its mechanical simplicity and by the number of moving parts. The fewer the number of moving parts, the simpler and the more reliable will be the assembly, since this reduces the possibility of jamming or seizing of parts and of the occurrence of nadirs on them; the number of seals is reduced (especially moving seals); the possibility of the influence of unnoticed defects during the manufacture of the parts on the valve's operation is reduced.

Precisely on the strength of these considerations disk-type cut-off valves, which have the simplest construction, are often employed as valves on LPREs.

The weight also has great significance for assemblies of LPREs (much greater than for assemblies of aircraft engines, not to mention other regions of technology). Of especially great importance is the weight for assemblies of engines of space vehicles - here the weight characteristics have truely decisive significance.

The weight of an assembly is closely connected with its hydraulic characteristics; the hydraulic resistance of the valve, as a rule, is the greater, the lower is its weight.

Let us pass on to an examination of specific (sometimes contradictory) requirements for assemblies, which determine their reliability.

Such basic requirements include:

1) the hermeticity of all the seals - both moving and stationary;

2) the stability of the opening and closing times;

3) the precision and dependability of triggerings;

4) the assurance of the necessary hydraulic characteristics;

5) smoothness of the process - no delays in the shifting of the movable system;

6) resistance of the materials in aggressive propellants;

7) explosion safety;

8) long storage time and operating service life (it is natural that the idea of the "service life" has significance only with respect to repeated-action assemblies);

9) strength of the construction;

10) manufacture ability; minimum cost.

These "autonomous requirements" may be analyzed independently of the engine operation, by independently examining each individually taken assembly, and a check for conformance to these requirements can be carried out during autonomous tests of the assembly.

1.3.1. AUTONOMOUS REQUIREMENTS

1.3.1.1. Hermeticity of the Seals

In the manufacture of propellant valves it is essential to provide for the reliable hermeticity of all seals.

In constructions of propellant pneumatic valves the following must be provided for:

a) the reliable separation of the air (controlling) cavity from the fluid (propellant) cavity;

b) the hermeticity of the locking mechanism during setting on its seat (the shut-off of propellant access must be guaranteed with the valve closed);

c) the complete hermeticity of flanged and threaded connections.

Known hermeticity of threaded connections in an assembly, as well as known hermeticity of welded joints, leading to the leakage of liquid propellant or its vapors outside, can cause fire onboard the flight vehicle.

The separation of the controlling cavity from the propellant cavity in pneumatic valves is ensured with the aid of rubber steel rings, other rings or bellows fields.

The hermeticity of the setting of the locking mechanism of the disk on the seat is achieved usually using soft end fields, manufactured from rubber, teflon or (more rarely) PTFCE and capron.

In explosive valves due to their one-time utilization and the widespread use of welded joints questions of providing hermeticity are not so very important.

However, it is essential here to eliminate the penetration of the combustion products of the explosive charge, which has a high temperature and pressure, outside; an insignificant leak (a trickle) of solid-reactant gases on the threaded joint of an explosive cartridge can lead to a hot spot on the thread and, as a consequence, to the extraction of the explosive charge from its seat. It is usually necessary to provide a seal on the cut element of an explosive valve (before the triggering of the explosive cartridge) and by the residue of the cut element (the

sealing collar) after its triggering. It is necessary to eliminate the possibility of the solid-reactant gases entering the propellant cavity of the explosive valve. Finally, it is necessary to ensure an adequately reliable seal of the locking mechanism in the valve housing.

In normally closed direct-action explosive valves the hermeticity of the locking mechanism after the triggering is ensured, as a rule, by the wedging of the metal valve into the seat of the housing. The valve itself has a small conicity while the conicity of the seat is ordinarily 2-4° smaller than that of the valve; sometimes the conicity in the housing is altered by the radius.

The normal force of the flash of the locking mechanism of the explosive valve after wedging amounts to 2-6T (with a cross sectional diameter of the valve 10-30 mm). This force depends on the charge of the explosive cartridge, the diameter of the piston, the width of the cut collar and so forth.

In explosive check valves, where the pressing force of the locking mechanism on the seat is not great, soft seals - PTFCE and teflon etc. - are frequently employed.

In normally closed direct-action explosive valves the separation of the cavities (before triggering) is accomplished by thin tie-plates, which are cut during the triggering of the explosive charges.

The hermeticity of the stationary connections of propellant pneumatic valves is ensured by packings manufactured, as a rule, from soft metal. For example, for oxidizer valves which handle liquid oxygen copper packings are employed, and for oxidizer valves handling nitric acid, as well as hydrogen peroxiue,

aluminum packings are employed. In explosive valves (one-time assemblies) welded joints are getting broader and broader use.

All of the sites of the seals of the assemblies must ensure the required hermeticity in the entire temperature range of the environment, provided by the technical specifications for operation, including after transportation, after and during vibration, after prolonged storage and so forth.

1.3.1.2. Opening and Closing Time Stability

To provide engine operating stability, an identity of the processes of starting and shutting down the stability of the time of triggering the automated assemblies is of the greatest significance. By the stability of operation of the automated assemblies of a LPRE we ordinarily mean constancy of the time interval from the moment of giving the command for the opening (or closing) of the propellant valve until the moment of the start of movement of the moving system of the assemblies, as well as the constancy of the moving time itself.

Of all the types of propellant valves the most stable in operation are direct-action explosive valves. The burn-up time of the explosive charge is so small (of the order of 0.001-0.002 sec), that the effect of various external factors on it is almost imperceptible. An increase in the burn-up time of the pyrotechnic composition even by two times has practically no influence on the operation of the assembly.

The operating stability of direct-action explosive valves and check devices in explosive valves of the check type does not require a more detailed examination.

The operation of check-type explosive valves is on the whole stable, in practice. The divergence in the triggering time is

caused by the structural design of the decelerators, which of necessity are employed to prevent hydraulic shocks, arising as a result of the rapid movement of the valve's locking mechanism. Designs of hydraulic brakes differ widely. The braking effect in the overwhelming majority of hydraulic braking devices is based on the retarded overflowing either of the propellant, or of some kind of viscous fluid from one cavity into another; the overflowing is executed either by clearances in the moving system, or by special discharge jets. Depending on the magnitude of the tolerances for the manufacture of the parts the clearances may change and, consequently, the braking effect may also be altered. If the viscosity of the propellant or the diametric clearances between the parts are significantly changed with the change in temperature, then this also will cause instability in the operation of the hydraulic braking device.

If hydraulic shocks can be tolerated in the system, then hydraulic braking devices are not necessary, and such an assembly will possess high stability. However, even when there are hydraulic braking devices in the final designs the divergence in the triggering kinds is, as a rule, small - up to 0.010-0.030 sec.

The greatest divergences in the opening (or closing) time, i.e., the greatest in stability, are produced by pneumatically controlled valves.

The opening (or closing) time of pneumatically controlled valves consists of the following:

a) the delay time for the start of movement with respect to the moment of command and

b) the time of movement of the moving system (the time of "pure" movement).

In the overwhelming majority of cases the duration of the movement itself is significantly less than the delayed time for the start of the movement. The delay in the start of movement of the valve's moving system (after the command to trigger the controlling EPV) is quite different for different types of assemblies and is not always stable for various samples of an assembly of one and the same type. This delay time depends on the volume of the controlling cavity, the hydraulic resistances of the communication lines of the controlling gas, the magnitude of the friction of the moving system, the spring forces, the temperature of the surrounding medium and so forth. As a result of this divergence, the instability of the opening (or closing) time of the valve is mainly determined by the duration of the delay for the start of movement of the moving system. The divergence in pneumatic valves, which handle cryogenic (low-boiling) oxidizers, is especially great.

1.3.1.3. Precision and Trouble-free Operation

This requirement means the elimination of the possibility of the hanging up of the moving system (delays during movement), the braking of parts, incomplete covering of the main line (during valve closing) or, on the other hand, incomplete opening.

The precision of valve operation is determined exclusively by the process of the change in the relationship of the forces acting on the moving system of the assembly- Therefore, in order to have reliability in the precise and reliable operation of the assembly, it is essential to determine the nature of the change in time of the forces which act during the process of triggering (opening, closing), which may be done both by the calculation method, as well as experimentally. These investigations are necessary, in order to be convinced that there is a sufficiently constant or monotonically increasing force acting in the direction of movement of the valve.

It is inadmissible that the resultant forces, applied to the moving system, be at any moment substantially reduced or change the direction. In pneumatic valves this requirement is ensured by the corresponding valve of pressure in the controlling cavity, by its volume and area, on which the controlling pressure acts; in explosive valves - by the correctness of the selection of the explosive cartridge charge and by the determination of the permissible value of the counteracting hydraulic shock.

Trouble-free operation of explosive valves is also determined by the continuity of the electrical circuit of the explosive cartridge. To increase the operational reliability of explosive valves duplication of the explosive cartridges or, what is more justified, duplication of the electrical circuit in the explosive cartridges is employed, since the possibility of a break in the circuit inside the cartridge or a defect in the external supply line is completely not excluded. Therefore, after engine assemblage, and if possible also before starting the engine, the continuity of the electrical circuits of the explosive cartridge should be checked; this is done with the aid of a very low current (a safety current), incapable of igniting the explosive composition in the explosive cartridge.

It must be kept in mind that at high temperatures spontaneous combustion of the explosive composition can occur without the supply of an electric current. Therefore the explosive cartridges must be protected in some way or another from the effects of high temperatures.

For explosive valves it is very important to eliminate the possibility of recoil of the moving system both as a result of hydraulic shock and as a result of other causes. To combat this phenomenon various structural measures are employed.

1.3.1.4. Hydraulic Characteristics

The hydraulic characteristics are one of the important parameters of propellant valves.

During continuous normal engine operating conditions the hydraulic resistance of an assembly can be the minimum (to reduce power loses of the turbopump assembly). During engine launch mode (preliminary stage of valve opening) the magnitude of the hydraulic resistance must conform strictly to the assigned value.

The hydraulic resistance of an assembly affects the value of the ratio of the components in the combustion chamber, especially the mode of the preliminary stage of engine operation.

In designing a new engine and its assemblies the magnitude of the hydraulic resistance in the preliminary and main stages is determined by calculation and is achieved by means of the selection of the appropriate configuration and dimensions of the flow-through section of the assembly. It is very important to know the actual magnitude of the hydraulic resistance of a valve both in the main stage of engine operation, with its complete opening, and especially during launch (preliminary stage).

The hydraulic resistance is, as a rule, determined for each assembly sample, for which special flow tests on the bench are carried out (see section 6.3). In certain cases it is sufficient to spotcheck the hydraulic resistance - with a successful design and adjusted manufacturing technology of the parts.

The hydraulic characteristics must not change during overhaul of the assemblies. However, overhauls of assemblies after a test flow are often necessary, since it is usually difficult to remove moisture from the assembly without disassembling after

a test flow. With welded valve design sometimes a test flow is conducted before the final welding of the parts, and final welding is done in units, that is, after the test flow the valve is partially disassembled. It is often necessary to repeatedly use pneumatic assemblies, which have already been used in an engine, after their disassembly (to remove propellant residues, and for flaw detection, etc.).

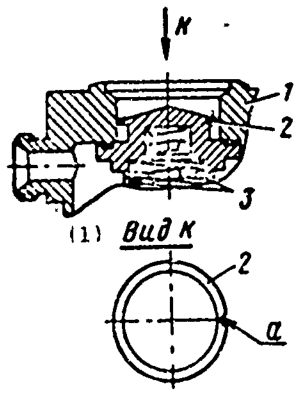

Fig. 1.12. An element of a propellant valve design: 1 - housing; 2 - head of the valve disk; 3 - spring; a - break end collar.
KEY: (1) View.

It is desirable to have a such a design, in which the hydraulic resistance in the preliminary stage of engine operation would be determined not by the value of the movement of the moving system, but either by the magnitude of the diametric clearances in the parts, or by the diameter of the opening holes, or by the triggering of special pilot valves - in all these cases the value of the resistance depends in no way on the reassembly of the unit.

Figure 1.12 shows a similar element of a valve design. The flow in the preliminary stage of engine operation is determined by the area of the opening (the notch) a in collar 2 of the disk. The magnitude of the opening is established experimentally for each valve during its flow test.

As an example of a design of another type we may use the oxygen valve depicted below in Fig. 2.1. It has a special pilot valve for operation in the preliminary stage.

A considerable influence is exerted in the magnitude of the hydrodynamic forces, i.e., on the magnitude of the hydraulic resistance, by the micro relief of the parts, especially of soft seals on valve seats; a difference in the seat profiles, in the shape of the seat identation, may lead to instability of the characteristics in various valve samples of one and the same type. The hydraulic characteristics are greatly affected by the edges on parts, lying in the flow-through section of the valve, by the fineness of the finishing of parts and by other "trifles" of such kind.

1.3.1.5. Smoothness of the Path

The requirement for smoothness of the path of the moving system is not decisive for ensuring the reliable operation of a fuel valve. For explosive valves the requirement for smoothness of the path, naturally, does not at all come into question.

The absence of smoothness of the movement of the locking mechanism of a pneumatic assembly irrefutably testifies either to the presence of significant friction forces, or to the poor centering of parts, or to the low quality of assemblage. Thus, a check of the smoothness of travel allows us to detect defects in the construction or manufacture of an assembly.

To ensure even travel measures are taken to reduce the shear and friction forces of the moving system of the assembly: mated parts are made from materials which possess different hardness of their rubbing surfaces, antifrictional protective coatings are used (most frequently, cadmium), the centering is improved and so forth.

The magnitude of the friction of the rubber seal rings is significantly affected by the quality of standardization of the rubber. Data on the influence of standardization on the value

of the friction force will be given in section 8.1. The coefficient of friction of the sealing rings depends also on the temperature at which they operate, and on the amount of lubrication. With an increase in the storage time of the assembly the coefficient of friction of sealing rings (and, of course, of other rubber parts) increases in connection with the change in the structure of the lubricant and with the aging of the rubber. A periodic check (during storage) of the smoothness of travel allows us to determine the character of this process.

The periodic determination of the pressure value (in the propellant or controlling cavities) of the start and of the end of movement of the valve's moving system serves for this same purpose - the investigation of the friction forces.

1.3.1.6. Endurance in Corrosive Media

The materials from which assemblies are manufactured must possess sufficient endurance with respect to the propellant components used, which are often very corrosive.

Propellant may be supplied to the intake valves long before the starting of the engine (this time is regulated by the operating technical specifications and depends on the layout and purpose of the flight vehicle); after the cessation operation propellant or its vapors may be left in the internal cavities of the assembly for a long time.

The most agressive with respect to the materials employed in the constructions of propellant valves are various types of oxidizers, although even certain of the employed types of fuels lead to the destruction or to defects of certain brands of rubber, polyvinyl chloride, and dissolve lubricants, cements and so forth.

To reduce the destructive effect of propellant components it follows to strive to reduce the contact time of the propellant and its vapors with the parts of the assembly, especially with parts containing nonmetallic materials (primarily rubber); to reduce the number of parts which come in contact with the propellant; and to reduce their area.

With an eye toward reducing the time of the contact of assemblies with the propellant, they are sometimes set up at the inlet to the engine special inlet diaphragms (see, for example, valves 15 and 16 shown in Fig. 1.1, and also valves 8, which are open before starting the engine either by a boost pressure, or by triggering explosive cartridges). Such diaphragms are manufactured from materials which are especially stable to the given propellant.

The selection of the construction materials which conform to the requirements of strength, manufacturability, and lightness and which possess sufficient endurance in the medium of the employed propellant is a very critical problem.

The type of propellant and its characteristics have a significant influence on the selection of the construction of the assembly and of the material of the parts.

Thus, the material of parts, which work in a medium of low-boiling propellants, must not be brittle. The use of carbon steels is not recommended here.

A considerable difference in the temperature, at which the check is made of the efficiency of pneumatic valves (shop conditions), from the operating temperature forces us to analyze the materials of mated parts with respect to the values of the coefficients of their linear expansion. One should pay attention

to the selection of the material of rubbing parts with an eye to eliminating the possibility of the occurrence of nadirs (both under shop conditions, as well as during operation).

When operating the engine on low-boiling propellants the use of commercial rubber parts is not recommended: the use of rubber seal rings is in this case completely inadmissible.

In pneumatic valves the propellant and the controlling cavities are separated by bellows manufactured from the appropriate materials.

The sealing of the locking mechanism is accomplished by means of seating a metal valve on a metal seat at high specific pressures. For valves working in a medium of liquid oxygen it is possible to employ seals made from teflon.

For the manufacture of the housing assemblies operating in a medium of liquid oxygen cast aluminum alloys are employed. Aluminum alloys, such as AV, AK8, V95, copper and copper alloys are used for parts of the construction.

For the manufacture of the bellows stainless steel Kh18N10T is frequently employed.

Copper and alloys containing copper are unusable for components which work in a medium of oxidizers based on nitric acid, since copper alloys are subjected to corrosion. Carbon steels are also unacceptable in this case.

The most suitable material for the manufacture of parts of the housings of the assemblies, which work in a nitric acid oxidizer medium, are aluminum alloys and stainless steels. The locking mechanisms are made from chrome-nickel steels of type 18-8. In this case, their short-time storage in the propellant

medium for the manufacture of the end seals it is possible to use special kinds of rubber, however this is undesirable. Fluoroplastic is very resistant in these propellants.

The separation of the controlling cavity of pneumatic valves from the propellant cavity is also accomplished with the aid of stainless bellows.

To reduce the probability of corrosion of parts of the assemblies anticorrosional coatings, for example, chromium, are used.

Aluminum alloys of type AMg7 and AV are used for the manufacture of the housings of assemblies which work in a hydrogen peroxide medium. Parts of these assemblies are manufactured from stainless steels (with limited contact time with the product), or (still better) - from aluminum alloys. The application of copper or alloys containing copper is inadmissible. Many parts are electroplated with an anticorrosional coating. For assemblies which work in a hydrogen peroxide medium it is especially important to maintain absolute cleanliness of the parts, the complete absence of dirt, which could be a catalyst and cause decomposition of the peroxide.

The separation of the controlling and the liquid cavities of pneumatic valves may be accomplished not only by bellows, but also using seals or rings, made from the appropriate brands of rubber. In locking mechanisms rubber seals are usually used.

Petroleum-based fuels ensure the possibility of a wide range of selection of structural material and assembly layouts. However, here a careful approach is required to the selection of the kind of rubber, since many brands of rubber break down or swell under the effect of this or that type of fuel.

Steel 50KhFA is ordinarily used as a material for the manufacture of the appropriate springs, which, as a rule, work under great stresses.

Information on materials, from which specific parts are manufactured, is shown in Chapter 2 - in the description of typical valve designs.

1.3.1.7. Explosion Safety

In the initial stage of finishing engines there sometimes occur explosions, the cause of which is this or that omission in the construction, the disregard of the rules of operation or assemblage of the propellant valves.

Highly active propellant components of LPREs, especially oxidizers, require very attentive and careful handling. The explosion safety of oxidizer valves is closely related to the type of lubricant employed, but not only to it. To eliminate the possibility of explosion inside the assembly, during assemblage, check routine tests and operation of the assembly, one must ensure a high degree of cleanliness not only of the valve parts themselves and of the working tool, but also of the surrounding facility (the work site, the bench, the test stand propellant distribution lines and so forth). Sterile cleanliness in all sections of the assemblage and testing of the assemblies is a strictly necessary condition.

In starting the engine oxidizer vapors may enter the main lines or the fuel assemblies, or on the other hand - fuel vapors may enter the main oxidizer lines, leading to an explosion. The reason for this may be the nonhermicity of one or of both propellant valves.

To eliminate the possibility of an explosion here during starting one or both of the main lines are usually blown out with an inert gas, usually nitrogen (in the system shown in Fig. 1.1 the blowing out is accomplished through check valve 25). Such purging prevents the accumulation of propellant components in the cavities of the propellant valves, and prevents the penetration of oxidizer vapors into the main fuel lines and so forth.

In designing valves intended for operation in a medium of more active oxidizers one should eliminate the possibility of the formation of chips or projecting edges at the moment of movement or the wedging of parts, since shavings in the presence of a spark (from impact) can be ignited. Residues of cement, lubricant or other similar substances are particularly dangerous at the sites of impact of parts.

The penetration of combustion products from explosive charges into a closed cavity, where the vapors of the active oxidizer are found, may present the dangerous threat of an explosion.

Certain metals, especially at high temperature (which may arise as a result of friction or the scoring of parts), have a catalytic effect on the propellant, causing its decomposition with an abrupt pressure increase; for example, copper and oxides of iron have this effect on hydrogen peroxide.

1.3.1.8. The Operational Service Life and the Storage Duration

For repeating valves the required service life of the triggering mechanisms is determined undoubtedly not by the number of triggerings during engine operation, but by the number of triggerings during the checks of the assembly's efficiency.

During manufacturing a pneumatic valve is subjected to repeated checking by the very worker who assembled it, and by the quality control representative of the department of the plant (these triggerings, occurring during the stage of initial assemblage, are not taken into consideration in calculating the service life of the triggering mechanisms; only triggerings after the assembly leaves the manufacturing shop are taken into consideration).

After installing the valve in the engine, to test the correctness of functioning of the various systems of the engine in the assembling shop, the assemblies are also subjected to multiple checking by the assemblers and controllers. After shipment of the engine from the plant, and after its installation in the flight vehicle the pneumatic valves of the engine are sometimes again tested for correct functioning. Thus, a rather significant number of triggerings can be gained from the total of these tests.

But what limits the necessary service life of a pneumatic valve, i.e., which part or unit restricts the permissible number of operations?

In those instances where the design contains a bellows, as a rule, the service life of the assembly is limited by the service life of this bellows. With an exceedingly great number of operations a crack may develop (most frequently at the end of corrugations); this is equivalent to the service failure of the entire assembly. In order to increase the service life of a bellows, attempts are made to decrease the linear deformation occurring on each corrugation during opening (closing) of the valve, and to avoid alternating loads, and the bellows are equipped with special bands and so forth[1].

[1]The operation of bellows seals is described in Chapter 3.

A fuel valve configuration which does not use bellows seals has a significantly greater service life than a valve design with bellows. In this case the permissible operational lifetime is in great measure determined by the quality and quantity of the grease, carried on the rubbing surfaces of the rubber seals, brings, etc.

With inadequate lubrication friction increases at the sites of the seals and the moving system (especially at low temperature) is shifted with greater forces; because of the increase in shear the opening and closing of the valve does not occur smoothly, but rather with jerks and so forth; consequently, the assembly design ceases to satisfy the technical requirements imposed on it. However, with the correct assembly of a pneumatic valve the required service life is, practically speaking, ensured without special difficulties. The degree of hermeticity of the locking mechanism of the valve in those cases where the seal is rubber or relatively soft plastic (for example, teflon), as a rule, also does not limit the operating service life, i.e., with the number of openings and closings of the assembly required during operation the hermeticity of the valve's locking mechanism remains practically unchanged.

Besides the service life, it is essential to ensure the required time of storage of the assembly in contact with the propellant, which has been poured into the inlet cavity. This requirement is also extended to normally closed explosive valves. However, the fulfillment of this requirement for explosive valves presents no problem; ordinarily the inlet cavity of a normally closed valve is restricted by a metal diaphragm or copper diaphragm, the material of which is resistant to the propellant.

For pneumatic valves, in which the propellant stored in the inlet cavity comes in contact with rubber seals, the time of storage in contact with the propellant is significantly restricted, since the rubber can swell, allow the propellant to pass, gets all the grease and so forth.

Also essential is the careful fitting of the end seals of the pneumatic assemblies, especially in cases where temperature fluctuations are possible; temperature fluctuations impede the operation of the seals, especially seals made from teflon.

For all assemblies without exception there is the imposed requirement to maintain the working efficiency after storage, the period of which is defined by the technical specifications.

The antifrictional properties of the lubricant deteriorate with the passage of time as a result of the change in its state; the smoothness of travel of the assembly becomes worse, the shearing forces increase, as a result of which the valve's opening and closing times may change.

As a result of the phenomenon of "aging" of the rubber with the passage of time, the elasticity of the seal rings and of the end packing seals worsens, which may lead to a loss in hermeticity; most strongly affected by the aging of rubber are the hermeticity of the seal rings under low pressures, the hermeticity of the jamming of rubber diaphragms, the hermeticity of packing seals and so forth.

A more detailed account of the phenomenon of the aging of rubber is given in section 3.1.6.

With normally closed or normally covered valves (in which the valve disk is pressed against a seat by the force of a spring) it should be taken into consideration that with the passage of

a certain time there arises the "adhesion" phenomenon of the rubber to the seat. This "adhesion" of the rubber is the stronger the longer the storage period (in the compressed state) and the higher the temperature during storage.

With a poor selection of the materials of the mated parts or with a low quality of the coatings after a certain time the corrosional breakdown of the metal may take place, especially under conditions of a moist atmosphere. This refers both to one-time valves as well as to repeating valves.

It is important to ensure the preservation of such nonmetallic materials as glues, cements, etc. Therefore, it is important that the manufactured articles employed in the assemblies and the supplied nonmetallic materials have verified warranty storage periods, in the course of which the maintenance (or a reduction within the permissible limits) of the basic properties of these manfactured articles and materials (strength, elasticity, etc.) is ensured.

1.3.1.9. Strength, Manufacturability, and Cost

Just as for any assemblies found on board a flight vehicle, for valves of a LPRE such an index as strength is of primary significance. During the entire given service life the parts of the valves of LPREs must not have residual deformations (with the exception of the soft seals), not to mention cracks, breakages or failures. All these defects are not even permitted for unimportant parts.

Considering the value of the weight characteristics decisive for the engine, the margin of strength of assemblies of a LPRE is taken to be low. The required strength and low weight must be insured by the creation of uniformly strong parts (avoiding

under stressed and over stressed sections) made from high-strength light materials. In connection with this the manufacturability of new materials, the study of methods of their mechanical working, welding, casting into shapes and so forth are of important significance.

A reduction of engine weight on the whole, is not of its individual assemblies, is one such prominent problem, and it must be solved first of all by the careful working out of the engine design, by the use of the most advantageous propellant components and by an optimum guidance system.

Thus, for engines of great thrust with one time starting a control system using explosive automatic devices ordinarily provides better weight characteristics and a pneumatic control system. However, this does not mean that an explosive assembly must of necessity be better than a pneumatic assembly (although in the majority of cases this is precisely the case).

A significant pressure variance, taking place during the triggering of explosive cartridges, prevents a reduction in weight of the parts (it forces us to make them with maximum strength); but since the pressure of the solid-reactant gases is usually absorbed by parts which are small in size, then the increase in the assembly's weight connected with this is small.

The essential strength of assemblies under the action of static loads from the propellant pressure, the controlling air, the acceleration of the flight vehicle and so forth is determined by calculation, and with the developed technology of parts manufacturing this is rather easily provided for. The greatest difficulty consists in ensuring strength of the assembly under dynamic loads - at the moment of triggering, during vibration and at the moment of shutting down the engine.

Problems of strength and reliability are closely connected with the technology of parts manufacturing. Experimentation shows that a change in technology, a seamingly even insignificant one, can influence the working capacity of the assemblies. Thus, the transitions to forging of a billet of a burst part of an explosive valve (if previously the part was machined from a rod) can lead to the nonactuation of the assembly; as a result of the strengthening of the part (due to the change in the direction of the grains of the metal) the value of the pressure, developed by the explosive cartridge, may be inadequate to break through the burst element. This example indicates that every change in technology requires a serious experimental test.

The creation of assemblies of LPREs, just as in any other field of new technology, gives birth to new technological processes and forces us to perfect old ones.

The production of automatic assemblies is today unthinkable without the utilization of the lost-wax process for the manufacture of valve housings, the utilization of automatic welding for housing parts, the use of new metals, alloys and synthetic materials. A high degree of purity and the degreasing of parts is achieved using ultrasound. In the making of long pin-holes (of the order of tenths of a millimeter), the electrospark machining method is employed. The electrospark machining method[1], as well as the electro impulse method, is used to process hard and high-temperature alloys. In preprocessing shops for the cutting of hard metal rods anode-mechanical cutting is used. However, the use of these new methods of machining do not at all mean that

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[1] The electrospark machining method allows us to make in metal of almost any hardness a given channel cross sectional configuration with an accuracy of the order of 0.02 mm and a purity of not less than ∇5 and eliminates the formation of projecting edges.

there is now the possibility of manufacturing any part, created by the imagination of the designer, and the problem of the "manufacturability" of the construction is not eliminated.

The manufacturability of a design on assembly means the possibility of its manufacture using modern equipment in conformance with strictly assigned (preliminarily worked out) machining conditions; this means that series production does not have to be made by the "skillful hands" of a single expert, and it might be made by any technically proficient, appropriately trained worker. Such an approach requires serious designer and manufacturer cooperation.

The valve design must eliminate the necessity for the adjustment or fitting of the proper parts; it must provide the possibility of the objective control of the quality of assembly and the possibility of removing moisture from the internal cavities after filling without the disassembly of the unit. Yet a reduction in the cost of manufacture of the assemblies must be accomplished first of all by increasing the reliability, which is also ensured by the manufacturability of the construction.

The manufacture of principally reliable assemblies allows us to sharply decrease the duration of machining, reduce the number of assembly samples, required for machining, and to decrease and simplify the test program, which in certain cases allows us to avoid the construction of special stands and facilities. All this leads to a reduction in expenses for the creation of an assembly and for the engine on the whole. Therefore, the structural development of various schemes of assemblies in the search for the most advantageous solution leads finally not to an increase in price and in the time for their manufacture, but on the contrary, to a reduction in price and to the acceleration of the times for producing reliable, finished constructions.

1.3.2. The Interaction of Assemblies During Engine Operation ("Layout Requirements")

It is impossible to formulate the "layout requirements" in a general form, or to determine the essential interaction between the assemblies of an engine. In each type of engine there must of necessity be ensured a definite, more or less exact, time sequence with the opening and closing of valves; it is essential that each assembly ensure the required time characteristics when interacting with other assemblies: if during autonomous tests the assembly satisfies those requirements, then under the influence of other assemblies its perimeters must not change in an adverse way.

The mutual operating influence of assemblies of an engine on one another occurs by the most diverse means - through the common conduits, the common EPV, the reducers and so forth. It is necessary to know how to determine and analyze this influence.

Pressure peaks and hydraulic impacts arising during the operation of a single assembly distort the operation of another assembly, and have peculiarities which cannot be understood by autonomous investigation of each individual assembly.

Commands to trigger the pneumatic valves, given by means of supplying a voltage to the EPV or by removing the voltage, during combined operation of two or more pneumatic assemblies may be distorted - drawn out in time, or delayed.

The most careful autonomous research on the efficiency of a valve sometimes proves to be inadequate to judge its operation in the engine.

As an example of the effect of the operation of one assembly on the functioning of another let us examine the case where, as

a result of the triggering of a liquid nitrogen valve, the operation of the hydrogen peroxide valve was disrupted.

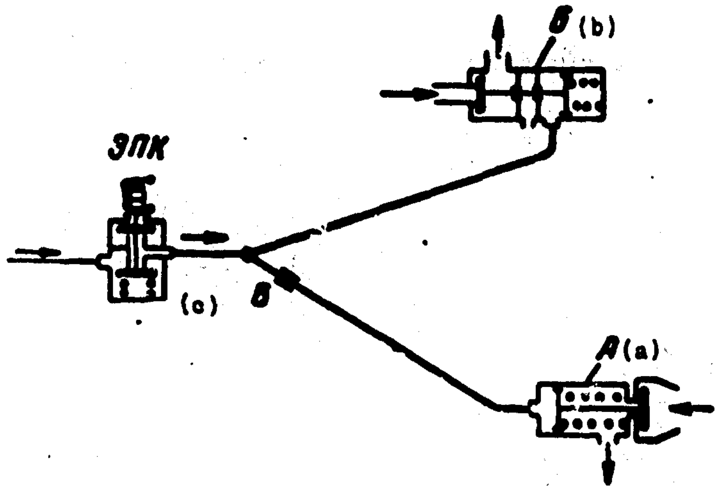

Fig. 1.13. Layout of pneumatic valve control: A - hydrogen valve; B - hydrogen peroxide valve; C - discharge jet.

The layout of these assemblies on the engine is shown in Fig. 1.13. (Other engine assemblies, which do not bear a direct relationship to the examined example, are not depicted in the diagram). The construction of nitrogen valve A is in principal similar to the construction of the assembly depicted in Fig. 2.5, while the construction of the hydrogen peroxide valve B is similar to the construction of the assembly shown in Fig. 2.7.

Both assemblies are normally closed, controlled by one common normally closed EPV with drainage.

During the autonomous tests of each assembly they operated completely reliably, opening and closing with precision. However, during the testing of the engine, when the EPV is de-energized and both valves must be closed, there were several instances of delays in the closing of valve B. Valve A closed at the proper time. If the closing time of valve B ordinarily amounts to 0.12-0.17 sec, during its nonstandard closing it amounted to 0.45-0.56 sec. Discharge jet C, depicted in Fig. 1.13, was at first not installed.

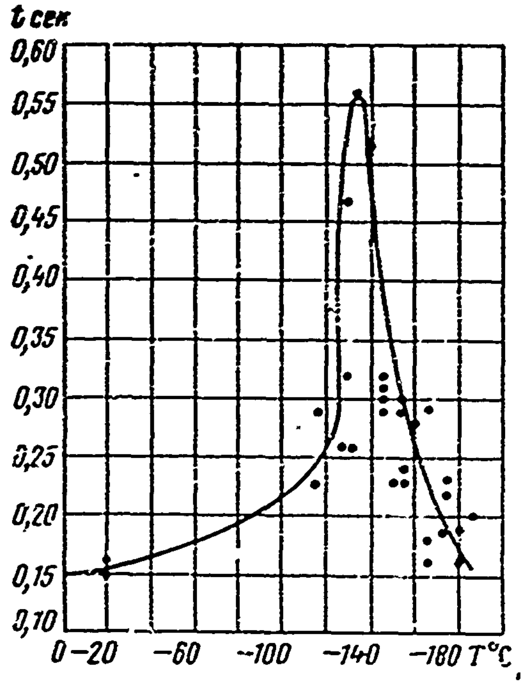

Fig. 1.14. Dependence of time t of closing of valve B on temperature T of the air in the controlling cavity of valve A.

An investigation showed that the reason for the delay is the reduction of the compressed air in the controlling cavity of valve A. During the shutoff of the EPV, when the pressure in the controlling cavities of both valves drops, in the controlling cavity of valve A boiling of the air begins, which has a chance to be condensed while valve A is in the open state. Because of this the drop in the controlling pressure is, of course, delayed. The time of release of the controlling air from the valves from a normal pressure value to a pressure equal to 1-2 at, instead of being 0.3-0.4 sec (with the temperature of the environment at +15-+20°C) amounted to 0.8-2.7 sec (with the presence of liquid nitrogen in valve A).

Why does such a delay in the release of the controlling air, which takes place constantly, not always affect the operation of valve B and not at all affect the operation of valve A?

The fact is that if the pressure in the controlling cavity, at which closing of the valve occurs, is higher than the pressure p_1, at which the boiling of air occurs, then this boiling does

not affect the process of closing, and if it is lower, then the closing of the valve is retarded. Since the controlling cavities of valves A and B are interconnected by a T-piece (see Fig. 1.13), then the controlling pressures in both valves at any moment in time are identical or almost identical (the EPV is a limiting cross section). However, the forces acting on valve A (the pressure of the product and the force of the spring) ensure its closing with a controlling pressure equal to 40-45 at greater than p_1. Therefore, the process of boiling of the air does not affect the operation of valve A, while valve B is closed at a pressure very close to the value p_1, and so even a small increase in the value of p_1, which can be caused by a whole slew of factors (the temperature of the valves, decrease in the heat exchange with the environment, increase in the initial pressure of the controlling air, and chiefly - by the reduction in the removal of liquified air during the initial period of depressurization), has a very significant influence on the moment of closing of valve B.

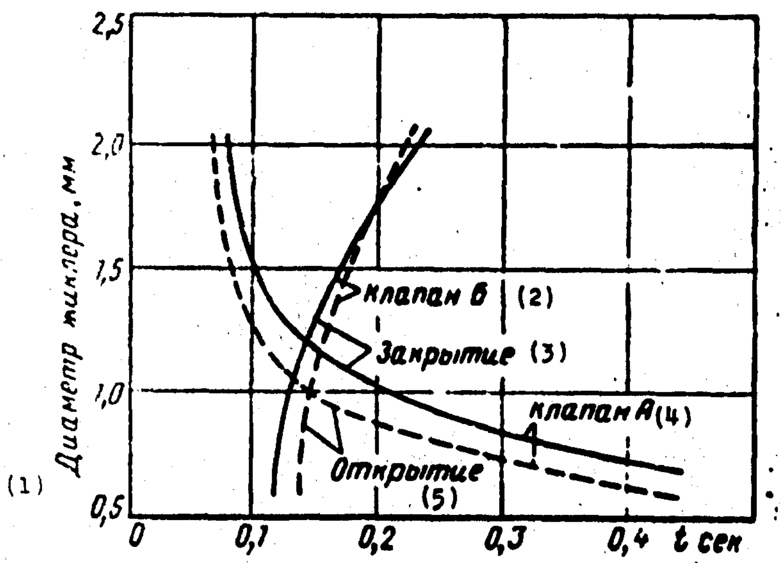

Fig. 1.15. Dependence of the opening time and the valve closing on the discharge jet diameter: - - - opening time; ——— closing time.
KEY: (1) Discharge jet diameter, mm; (2) Valve B; (3) Closing; (4) Valve A; (5) Opening.

Investigations have shown that the actual time of discharge of the controlling pressure from valve B depends on its temperature. Figure 1.14 shows the experimental data on the opening time for valve B as a function of the air temperature in the controlling cavity of valve A. The temperature of the controlling air depends mainly on the time for the flowing of the liquid nitrogen through valve A, i.e., on the duration of the start. The instances of the delay in the opening of valve B were the results of the action of these factors.

In order to eliminate the influence of the operation of valve A on the operation of valve B in the controlling pressure conduit discharge jet C was set up (see Fig. 1.13). The presence of a discharge jet retards the release of air from valve A, so that valve B manages to be closed until the moment of the start of boiling of the air in valve A.

Figure 1.15 gives the experimental curves for the change in the opening and closing times of the valves as a function of the discharge diameter. The smaller the discharge jet, the more quickly valve B closes, but the more slowly valve A opens when the EPV is switched on. The final value of the discharge jet diameter was selected on the basis of an acceptable value for the opening time of valve A, i.e., the time for the rise in pressure in the controlling cavity of the valve when the EPV was switched on. This example convincingly shows that the autonomous adjustment of the assembly is still in insufficient condition for its successful operation on the engine.

The assurance of the required engine starting conditions sometimes requires a change - acceleration or retarding - in the opening of this or that valve. And even if the process of opening of a valve was definite and stabilized during autonomous adjustment of the assembly, it is sometimes necessary to vary

it to ensure normal engine operation. This change may be accomplished by introducing certain structural changes into the given assembly or into the assemblies and connection lines related to it.

CHAPTER 2

A DESCRIPTION OF TYPICAL DESIGNS OF PROPELLANT VALVES

2.1. REPEATING VALVES

All the below-described repeat-action propellant assemblies, with the exception of the electric valve shown below in Fig. 2.14, are pneumatic valves, controlled by compressed air.

The main propellant valve (the oxidizer valve), represented in Fig. 2.1, has three fixed working positions, or three operating modes:

- closed;
- open for operation in the preliminary stage mode;
- open for operation in the main stage mode.

These working positions are provided by two individual locking mechanisms and two independent valves:

a) the main valve, or the valve of the main stage, 2 (see Fig. 2.1);

b) the small pilot valve, or the preliminary state valve, 1, which is mounted in the basic valve 2.

Propellant enters the assembly through the central flange. The discharge of propellant is accomplished through six branch connections, evenly arranged on the lateral surface of housing 3.

Figure 2.1 shows the assembly without pressure in the propellant (liquid) cavity A, or in the controlling cavity B. Here main valve 2 is closed, while prestage valve 1 is open. Thus, with the absence of air pressure in the controlling cavity the propellant may flow through the assembly; this position, depicted in Fig. 2.1, corresponds to engine operation in the preliminary stage.

When controlling pressure is supplied to connection 14 the air enters inside bushing 25 and compresses small bellows 5, overcoming the force of spring 13; now valve of preliminary stage 1 seats itself in the seat formed in the disk of valve 2. But the controlling pressure p_y, closing valve 1, at the same time, presses valve 2 into its seat in housing 3.

The hermeticity of the seal of valves 1 and 2 at the site of their seating is obtained by the use of rubber rings, vulcanized into the valve body.

The force for pressing the disk of valve 2 onto the seat in the presence of controlling air is determined by pressure p_y (acting on the area of the circle, formed by the effective diameter of main bellows 6) and by the force of spring 23. The force for pressing the pilot valve consists of the difference in forces from the pressure p_y, acting on the effective area of the small bellows 5 and compressing the bellows, and the force of spring 13, stretching the bellows (the area of screw 11 may be disregarded).

Thus, when compressed air is supplied to the controlling cavity of the assembly the latter is completely closed, i.e., the valves of the main and preliminary stages are closed. This is the situation until the starting of the engine. The propellant pressure in front of the valve p_T is small - it consists of the pressure of the column above the valve and the boost pressure in the tank. The forces acting on valves 1 and 2 reduce to a minimum the possibility of leakage of the propellant through the valve seals.

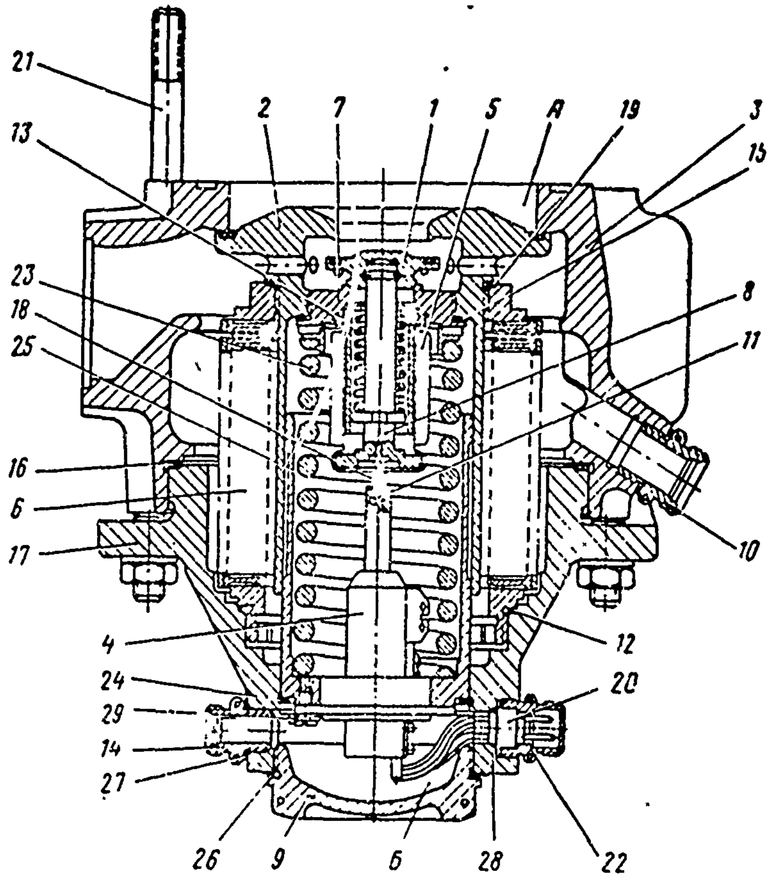

Fig. 2.1. Main oxidizer valve: A - liquid cavity; Б - controlling cavity; 1 - prestage valve; 2 - main valve; 3 - housing; 4 - switch; 5 - small bellows; 6 - main bellows; 7 - plug; 8 - rod; 9 - protective cap; 10, 14, 22 - connection tube; 11 - adjustment screw; 12, 15, 16, 19, 26, 27, 28 - seals; 13 - small spring; 17 - cover; 18 - spring of switch; 20 - block; 21 - pin; 23 - spring; 24 - stop ring; 25 - bushing; 28 - base; 29 - screw.

To go into prestage mode the air from the controlling cavity of the valve with the aid of the electropneumatic valve (EPV) is discharged into the atmosphere. Then under the action of the pressure of the propellant and of the force of spring 13 bellows 5 begins to be drawn out, and valve 1 opens. The travel of valve 1 determines the magnitude of the extension of bellows 5. Valve 1 begins to open a little before the controlling pressure falls to

atmospheric. Main valve 2 remains closed, since the propellant pressure is insufficient to overcome the force of spring 23.

To control the opening of the valve, so-called switch 4 (see Fig. 2.1) is added to the design of the assembly. With the movement of valve 1, fastened on rod 8, the sleeve of the switch moves; the sleeve is connected with rod 8 by means of spring 18 and adjustment screw 11, which is screwed into the base of small bellows 5. The sleeve in its movement closes the contacts of switch 4 and thereby gives the electrical signal, that the valve is opening for operation on prestage mode. With the aid of the adjustment screw (not shown in Fig. 2.1) the switch is regulated in such a way, that the contacts close with a definite opening value of the valve on prestage mode, for instance, with travel within the limits of 2-3 mm.

The propellant flow during opening of the assembly for operation on prestage mode is determined by the diameter of the openings in the body of valve 2.

With the activation of the main stage the propellant pressure rises abruptly, and valve 2 moves (compressing spring 23 and bellows 6) to detent with its lower face in the face of cover 17 (see Fig. 2.1). In this position the assembly is open for operation on main stage mode; valve 2 is completely open, valve 1 has not changed its position relative to valve 2.

It is important that the propellant pressure buildup occur according to the law required to provide the correct occurrence of the transient mode. During the propellant pressure buildup with valve 2 closed the drop in pressure at the valve will be very small (the flow passage cross-sectional areas are small). However, as soon as valve 2 is open slightly, the pressure drop is reduced and main spring 23 can again close the valve; then the pressure drop increases and the valve opens again. Such

an unstable operating regime will take place until the force from the propellant pressure, acting on bellows 6, exceeds the force of compressed spring 23.

It is natural that the opening and closing of the valve causes a corresponding increase or decrease in the propellant flow rate.

The hydraulic resistance of an assembly completely open for operation on main-stage mode must be minimal, so as not to make it necessary to increase the required pressure for supplying the components. On this basis the flow-through cross-sectional area and the magnitude of travel of main valve 2 are selected.

Before shutting down the engine the pressure at the inlet to the propellant valve is reduced. However, this reduction in pressure cannot be so great, that spring 23 begins to close valve 2.

With the complete shutting down of the engine the valve is closed, for which compressed air is fed into the controlling cavity. The air pressure, overcoming the propellant pressure in the liquid cavity, expands bellows 6, squeezing valve 2 onto its seat; valve 1 is also seated simultaneously on the seat in the disk of valve 2.

The controlling air pressure at which valve 2 begins to close is determined by the propellant pressure. In nominal mode the valve cannot close while compressed air is supplied; the force of the controlling air and of spring 23 is inadequate to overcome the propellant pressure.

The forces acting on the moving system of the assembly at any position and the pressure values at the start of movement are easily calculated. The pressure of the start of opening for

operation on preliminary and main-stage modes, as well as the pressure for the start of closing, can be determined experimentally when testing the valve.

Let us stop on the structural peculiarities of the valve. The elements which separate the controlling cavities from the oxygen cavity are bellows 5 and 6 (see Fig. 2.1), made from stainless steel Kh18N10T, welded to the mounting, and packing seals 12, 15, 19, made from soft aluminum.

The bellows are single-layer, with a wall thickness of 0.2 mm (main bellows 6) and 0.22 mm (bellows 5 of the preliminary stage). The compression of each corrugation of the bellows of the preliminary stage during valve operation comprises 0.5-0.7 mm; the deformation of each corrugation of main bellows 6 comprises 1.6-1.9 mm.

Under working conditions the bellows are subjected to compression by external pressure (bellows 6 - by the propellant pressure; bellows 5 - by air pressure) and to elongation by internal pressure. The main bellows is shown in Fig. 2.2.

To increase the stability and resistivity to external pressure inside each corrugation of both bellows there is a split snap ring 2, made from OVS (oxidation-reduction medium) wire (see Fig. 2.2).

The diameter of the wire of the ring of the main bellows is 2.6 mm, and in the preliminary stage bellows - 1.5 mm. So that the ring does not crease the wall of the bellows during compression of the bellows at its end, in the main bellows on one of the ends of the split ring there is welded a so-called "flag" - cover 3 (see Fig. 2.2). The other end of ring 2 is inserted inside cover 3 when the ring is installed in the bellows. (Cross section A - A in Fig. 2.2 shows the ring up to its insertion into the

corrugation). In the small bellows it is, in practice, impossible to set up such a "flag," therefore an especially careful rounding of the faces of the ring is provided for in it.

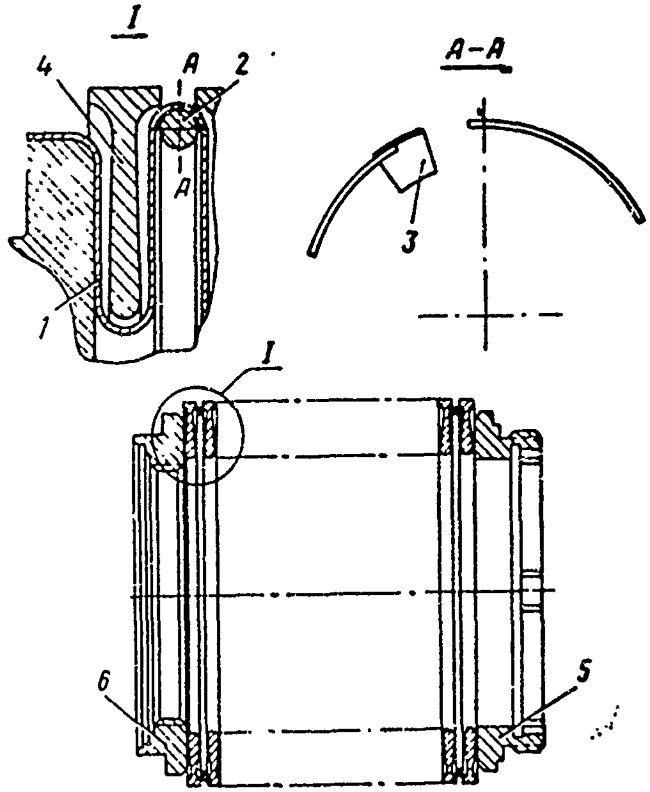

Fig. 2.2. Bellows: 1 - bellows connection; 2 - split ring; 3 - cover ("flag"); 4 - outer ring; 5 - casing; 6 - nut.

To prevent the inflation of the main bellows by the pressure of the controlling air between the corrugations of bellows connection 1 steel snap rings 4 are inserted. These rings give rigidity to the bellows and limit the amount of its axial compression. Rings 4 are installed during the process of manufacture of the bellows connection 1 - in the corrugation process. Sometimes in place of whole rings two aluminum half-rings are used, joined by a band, pressed over the outer diameter of the half-rings. Such

a design permits the installation of the rings not during the process of manufacture of the bellows connection, but after it.

Housing 3 and cover 17 (see Fig. 2.1) are manufactured from aluminum alloy type AL4. The strength of the housing walls is checked by a hydraulic press, and the porosity of the casting - by pneumatic testing.

If the valve is used for handling liquid oxygen, then pins 21, fastening the valve to the pump, and the pins, joining cover 17 and housing 3, are also manufactured from aluminum alloy (in this case it is desirable to use the strongest alloys of type AK8, V95 and others). If, in this case, the pins are made of steel and are screwed tightly into the aluminum housing, then with the decrease in temperature, during cooling by the liquid oxygen, cracks will appear in the housing as a result of the difference in the coefficients of linear expansion of the steel and of the aluminum.

Valve 2 and bushing 25 are manufactured from aluminum alloy AV. The application of aluminum is advisable in connection with the fact that, besides the low weight and the assurance of constancy of the diametric clearances (because of the identical coefficients of linear expansion), a durable fastening of the rubber seal ring to the metal is achieved in this way. (Rubber adheres poorly to alloys and stainless steel, and under conditions of low temperatures it can separate from the metal). Valve 1 is also manufactured from aluminum alloy.

The height of the seat for valve 1 (in the body of the disk of valve 2) is $0.5^{+0.2}$ mm; the seat is formed with a radius of 0.4 mm. The height of the seat in housing 3 for valve 2 is $0.7^{+0.2}$ mm, and the radius of the seat is also equal to 0.4 mm.

Paronite seal 16 is used to seal housing 3 and cover 17. The sealing of the threaded connection of the connection tubes with the housing is achieved using soft aluminum gaskets. However, in the case of operation of an assembly using liquid oxygen the use of paronite or aluminum gaskets is not recommended for sealing joints, which handle this product under high pressures.

For the sealing of threaded and flanged joints which handle oxygen it is desirable to use thin copper cadmium-plated gaskets installed in a closed volume. Copper has a high self-ignition temperature and high thermal conductivity. Cadmium, besides keeping copper from corrosion, reduces friction, since it has unique lubricating properties. This feature is very valuable for joints handling liquid and gaseous oxygen under high pressure where the application of lubrication is limited. Only in individual cases, for example when installing gaskets or when screwing in pins on a tight thread, is it permissible to rub with a chamois, impregnated with lubricant, parts which have relative movement only during assembly, and not during operation.

The main fuel valve, depicted in Fig. 2.3, is a normally covered pneumatic valve of the nonbalanced type, and of direct action.

The valve has only two fixed working positions: a) completely closed; b) completely open.

Figure 2.3 depicts assembly with no pressure both in the liquid (propellant) cavity Д, and in the controlling air cavity E; here the force of spring 7 presses the locking mechanism - the bushing of valve 2 against its seat in housing 5.

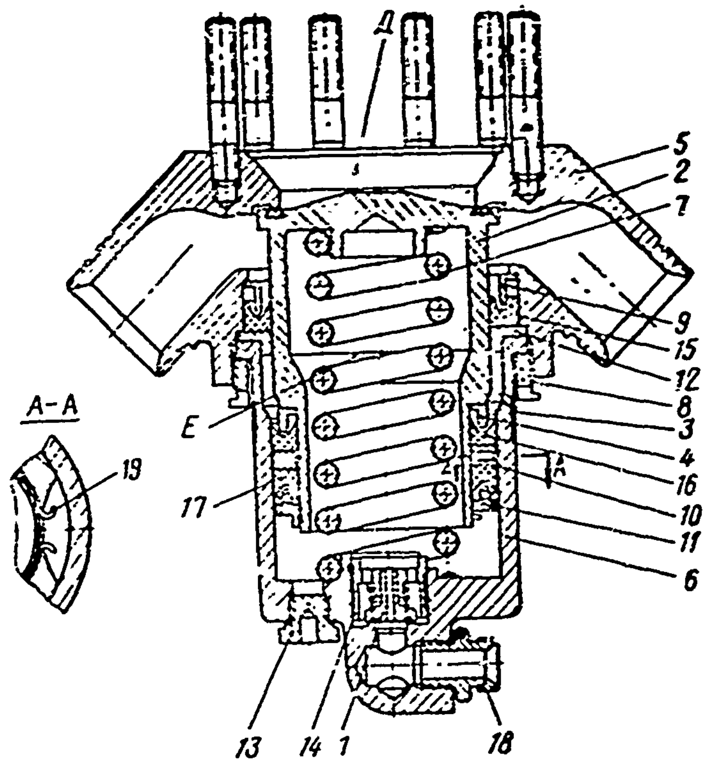

Fig. 2.3. Main fuel valve: Д - liquid cavity (inlet); E - controlling cavity; 1 - check valve; 2 - bushing; 3 - rubber ring; 4, 9, 10, 11, 12 - rings; 5 - housing; 6 - cover; 7 - spring; 8 - connector nut; 13 - plug; 14 - check valve housing; 15, 16, 17 - sealing rings; 18 - connector tube; 19 - gasket.

When controlling pressure p_y is applied to branch connector 18 the air passes inside the bushing of valve 2; the pressure of the compressed air (together with spring 7) creates significant force, acting on the valve disk, and ensuring an airtight seal against the seat of the housing. The valve is found in this position until the start of engine operation.

The pressure p_T of the fuel in front of the valve during this period consists of the pressure of the column of fuel above the valve and the tank pressurization pressure. It should be noted that even in the case where there is no controlling pressure, the force of spring 7 is sufficient to provide hermeticity at this fuel pressure; thus, the layout with a normally covered valve increases the safety of operation.

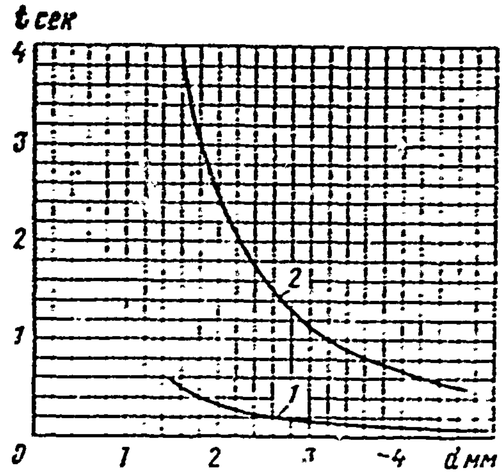

Fig. 2.4. Dependence of the time of valve opening on the bore diameter d in the rod of the check valve: 1 - time from the command to trigger the EPV to the start of opening of the propellant valve; 2 - time from the command to trigger the EPV to complete opening of the valve.
сек = s

To open the valve, air from the controlling cavity is discharged through the EPV into the atmosphere. Here check valve 1 presses against its seat and the rate of change (the reduction) in the air pressure is determined by the size of the central opening in the rod of valve 1. As a result of the drop in the air pressure in the controlling cavity and the rise in the fuel pressure at the start of the engine the valve begins to open, overcoming the force of spring 7, the resistance of the controlling air and the friction of the seals. When the valve is open slightly and the fuel begins to enter the engine, the pressure at the inlet may be reduced abruptly (the nature of the change in pressure is determined by the rate of speed of the turbopump assembly). However, since the value of the area, on which the fuel pressure acts, is increased (the outer diameter of bushing 2 of the valve, which is sealed by sealing 15, is greater than the diameter of the seat in the housing), and as a result of the fact that the controlling pressure continues to be supplied, the movement of bushing 2 of the valve will be continued, consequently no oscillations of the valve will be observed under actual operating conditions.

The movement of bushing 2 of the valve will proceed until the moment of its detent into cover 6. This will also be the position of complete opening.

The moment of the start of the opening of valve bushing 2 (relative to the command to the EPV) and the speed of opening can be regulated to a certain degree by the magnitude of the diameter of the opening in the body of valve 1.

Figure 2.4 presents the experimentally determined curve of the dependence of the time of opening of the valve (relative to the moment of switching on the EPV) on the bore diameter d in the rod of valve 1 (see Fig. 2.3).

To close the valve, air is supplied to the controlling cavity with the aid of the EPV. If the discharge of air through the opening in the body of valve 1 occurred relatively slowly, then with the filling of the cavity the bore diameter is not important, because check valve 1 is opened and the air pressure increase occurs abruptly; under the action of the controlling pressure and spring 7 the valve closes, overcoming the resistance of the flow of liquid and the friction in the seal rings.

Let us look at the structural peculiarities of the fuel valve. The sealing of the locking mechanism is accomplished by a ring made from teflon (GOST 10007-62[1]) and pressed into the body of valve bushing 2. For better fastening of the ring the groove in the valve body has a dovetail shape[1]. The seat in housing 5 is wide (the width of the band of the seat equals 2.8 mm), the height is 0.5 mm, and is of the open type. The maximum specific pressure on the fluoroplastic does not exceed 460 kgf/cm^2.

The separation of the liquid cavity Д from the controlling [illegible] is ensured by rubber seal rings 15 and 17. Since [illegible]

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[1] [illegible] in more detail the fastening of parts [illegible]

rubber under the action of the fuel on it swells and is capable of (in small quantities) passing fuel through it, there is a second, backup seal ring 16, fastened to valve bushing 2. Vent to the atmosphere is provided between seal rings 15 and 16. The vent openings are covered with rubber ring 3 to prevent dust and moisture from getting in.

The hermeticity of the air (controlling) cavity is ensured by seal ring 17, installed on valve bushing 2.

Seal rings 16 and 17 are fastened with the aid of rings 10 and 11, which are secured on valve bushing 2 using a special lock. Fastening is accomplished by soft steel locking wire 19 having a diameter of 1.2 mm, wound into the combined half-grooves, made in valve bushing 2 and in rings 10 and 11, respectively. The ends of the wire are bent in the specially made cuts in rings 10 and 11 (see cross section A-A in Fig. 2.3) Because of the small diametric clearances between valve bushing 2 and rings 10 and 11 (a third-class running fit) such a connection ensures a reliable fastening of the parts; stresses (for shear and warping) are small here. Disassembly of the connection is very simple. For this it is sufficient to pull the wire from the groove with pliers, after first unbending (or breaking off) one end of it. The connection allows us to make many reassemblies (of course, only the wire is replaced). Such a small-sized and very reliable lock has found widespread application in diverse propellant assemblies. Thus, for example, in the valve design under consideration the fastening of cover 6 to housing 14 of the check valve is manufactured similarly.

Cover 6 is fastened to housing 5 by connector nut 8 with a male thread. This same nut with the aid of rings 12 and 9 fastens seal ring 15. Between the seal ring and the rings there must be an axial clearance of 0.3-0.6 mm.

The threaded opening in cover 6, filled by plug 13, permits the possibility of installing a motion sensor for oscillograph recording of the character and magnitude of travel of the valve.

The sealing of check valve 1 is accomplished with a ring, made from rubber, vulcanized into the valve body.

When the units are assembled the rubbing parts are lubricated with a very thin layer of lubricant. It is necessary to lubricate the rubber seal rings very carefully. Both too much and too little lubricant can have a negative effect on the efficiency and hermeticity of the seal rings. Therefore, the amount of lubricant put on the parts during assembly is strictly regulated by technology[1].

The propellant valve, shown in Fig. 2.5, is designed to handle low-boiling fluid. It is a normally closed pneumatic nonbalanced-type valve with return action.

The liquid enters connector tube 13, situated along the axis of the assembly, and exits through the lateral connection tube, which is manufactured as one piece with housing 11.

The assembly has only two fixed positions:

a) completely closed;

b) completely open.

Figure 2.5 shows the assembly with no pressure in the controlling cavity. When pressure is supplied to the inlet of the assembly the pressure of the disk of valve 3 against the seat becomes

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[1]This refers not only to the given assembly, but to all cases where rubber seal rings are employed.

greater and greater. Opening of the assembly is accomplished by supplying compressed gas (air) to controlling cavity 5 through connector E of bushing 12. With the increase in pressure in cavity 5 to a value p_{y1} bellows 1 becomes smaller and base 2 forcibly moves valve 3, compressing spring 4. In connection with the increase in the volume of the controlling cavity the rise in the controlling pressure is retarded. Compression of the bellows will occur in proportion to the increase in pressure up to value p_{y2}, until base 2 is steeped in guide 6; a further increase in the pressure of the compressed air in cavity 5 will be absorbed to guide 6. This will also correspond to the position of complete opening of the valve.

If at the inlet to the assembly (in connection to 13) before compressed air was supplied the liquid was under pressure p_{T0}, then the force of the compressed air, naturally, must overcome at the first moment the entire force from the pressure of the liquid, acting on the seat area. When the assembly is opened the resistance force of the liquid is reduced and only now is the differential pressure Δp on both sides of the disk overcome.

Thus, the value p_{y1} depends not only on the force of spring 4 and the rigidity of bellows 1, but also on the initial value of the pressure p_{T0} and area F of the valve seat.

A diagram of the change in the air pressure, the pressure drop of the propellant and the amount of movement of the valve h during opening is shown in Fig. 2.6a. The change shown by the dashed line Δp_T occurs during the pressure increase in front of the assembly above value p_{T0} as a result of the increase in the flow rate.

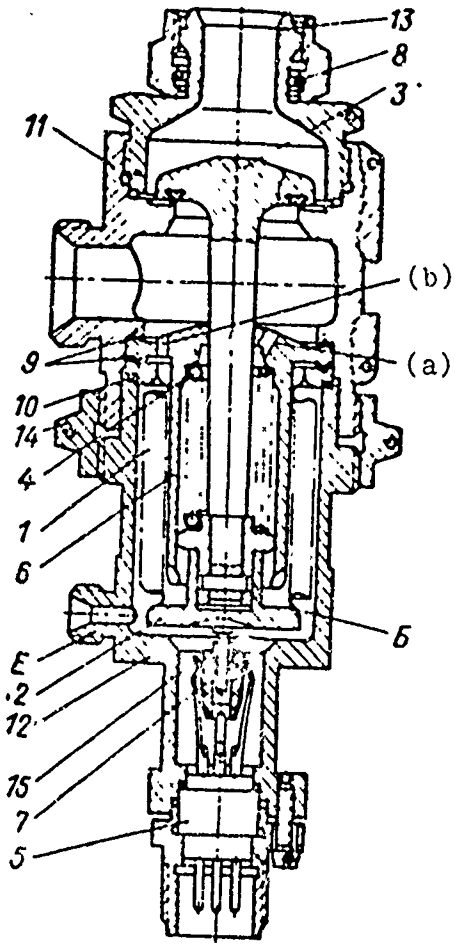

Fig. 2.5. Propellant valve: Б - controlling cavity; E - connection tube; a, b - bores; 1 - bellows; 2 - base; 3 - valve; 4 - spring; 5 - block; 6 - guide; 7 - bushing; 8 - stop ring; 9, 10 - gaskets; 11 - housing; 12 - bushing; 13 - connection tube; 14 - coupling nut; 15 - contact plates.

To close the valve, the pressure from cavity Б (see Fig. 2.5) is discharged through connection tube E. When the pressure falls to some value p_{y3}, under the action of the force of spring 4 and pressure drop Δp_T on both sides of the disk, valve 3 begins to be covered. In proportion to the closing the pressure drop Δp_T increases, because the flow through cross section in the assembly is reduced. Since the space inside bellows 1 is connected by holes a and b with the liquid cavity, the pressure drop Δp_T will affect the entire area of the valve seat. The rate of movement of valve 3 is determined by the rate of flow of the liquid and the speed of the decrease in pressure in the controlling cavity.

Figure 2.6b shows the change in the parameters of the valve during closing; it is assumed that the propellant pressure p_{TK} in front of the valve during the movement of the movable system of the assembly changes slightly. Such a design of the assembly is also suitable for handling ordinary liquid propellants.

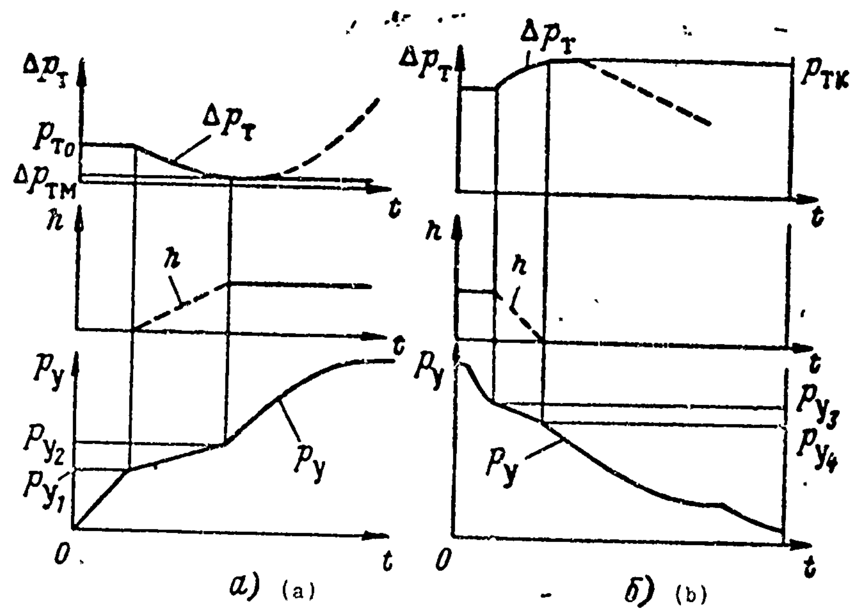

Fig. 2.6. The process of the change in the parameters of a propellant valve: a - valve opening; b - valve closing; Point "0" - the moment of giving the command to trigger the EPV.

To get the signal for the start of opening or for the end of closing of the valve there is a special electrical signal indicator, put together in the following way (see Fig. 2.5). On the threaded stud, made in one piece with base 2, is screwed textolite bushing 7, onto one half the length of which a copper ring is pressed. After the pressing of the ring the bushing is adjusted in such a way that the surfaces of the textolite and of the metal ring are flush; then the ring is covered with a thin layer of silver. On the small flange of bushing 12 plug 5 with four insulated contacts is placed. Silver-plated elastic contact plates 15 are soldered to each contact. Two independent electrical circuits

are supplied to the block with the aid of a plug (not shown in Fig. 2.5).

When the valve moves the contact plates at first slide along the textolite (the negative allowance of each plate should be 0.5-0.8 mm), and then, after traveling 1.5-2.5 mm, they come in contact with the ring and close the electrical circuit giving the signal to transfer the valve.

Because of the possibility of the deformation of the bellows and, as a result of it, the misalignment of the base and of the contact bushing, and also because of the deformation of contact plates 15, a backup is provided for the electrical signal, for which two independent electrical circuits are inserted.

The moment of closing of the contacts is regulated by the position of the insert by using screws of various thickness. The insert is fastened in the necessary position by cement BF (care should be taken so that the glue does not reach the surface of the ring and does not come in contact with plate 15).

Let us stop on the design peculiarities of the assembly. The seal of the valve is teflon, pressed into the disk of valve 3. The seat is of the open type, with a height of $0.7_{-0.1}$ mm, and a width of $0.8_{-0.2}$ mm. The radius at the top of the seat equals $0.4_{-0.1}$ mm, and at the base - 0.2 mm. Housing 11 and the valve 3 itself is manufactured from aluminum alloy.

The bellows is seamless, made from stainless steel Kh18N10T with a thickness of 0.2 mm. Inside the corrugations of the bellows wire rings are installed for rigidity. The bellows is welded by seam welding to the base and to the ring, manufactured from steel 1Kh18N9T. The connector nut is fastened to connector tube 13 using stop ring 8, inserted in the body of the nut after putting the latter on the output nipple of connector tube 13.

Bushing 12 is tightened to housing 11 using aluminum nut 14, which has on its side opposite the bushing a left-handed thread, and on the other side - a right-handed one.

The sealing of bellows 1 with bushing 12 is ensured by aluminum gasket 10, which has a thickness of 1.5 mm.

The hermeticity of the joints of guide 6 with housing 11 and the ring of bellows 1 is ensured by thin (0.5 mm thick) aluminum gasket 9. In order to facilitate obtaining the hermeticity, at the end of the ring of bellows 1 annular slots (grooves), 0.5 $\pm$ 0.1 mm thick, are made, and on the adjacent ends of guide 6 - corresponding projections with a height of 0.5 mm are also made.

Contact plates 15 are manufactured from strip phosphorus bronze of brand BrOF 6.5-0.15-T. Four-contact block 5 is made from plastic. The seal of its joint with the bushing is ensured by a soft aluminum gasket.

When the assembly is used to handle a low-boiling liquid it is desirable to set it up in such a way that connector tube E of bushing 12 for supplying compressed air is at the upper most point of the control cavity (in any case, not at the lowest); such a requirement has been caused by fear, that with the condensation of moisture, which is in the compressed air, or which enters the control cavity through the EPV from the atmosphere, icing or the complete sealing of the flow passage cross-sectional area in the connector tube will occur, leading to breakdowns in the operation of the assembly.

The normally closed valve, depicted in Fig. 2.7, is a nonbalanced-type, return action assembly. It has only two fixed positions - "closed" and "open".

Figure 2.7 shows the valve without pressure in the controlling cavity A, and in the closed position. The supplying of working pressure to the inlet to the valve does not change the position of the parts, since the magnitude of the force, created by springs 10 and 11, ensures the pressing of valve 1 to its seat in connector tube 14 (the force to valve 1 is transmitted through connector tube 7).

In supplying the controlling pressure to connector tube 16 of housing 2 springs 10 and 11 are compressed until bushing 7 reaches detent 9. If propellant is not supplied to connector tube 14, then valve 1 remains compressed against its seat by weak spring 3. Even when a small pressure exists, the liquid column will open valve 1 and the flow of propellant through the valve will commence. The force of spring 3 will be balanced by the magnitude of the pressure differential on the valve (the cavity inside the valve is connected with the outlet cavity of the assembly by bores a and b).

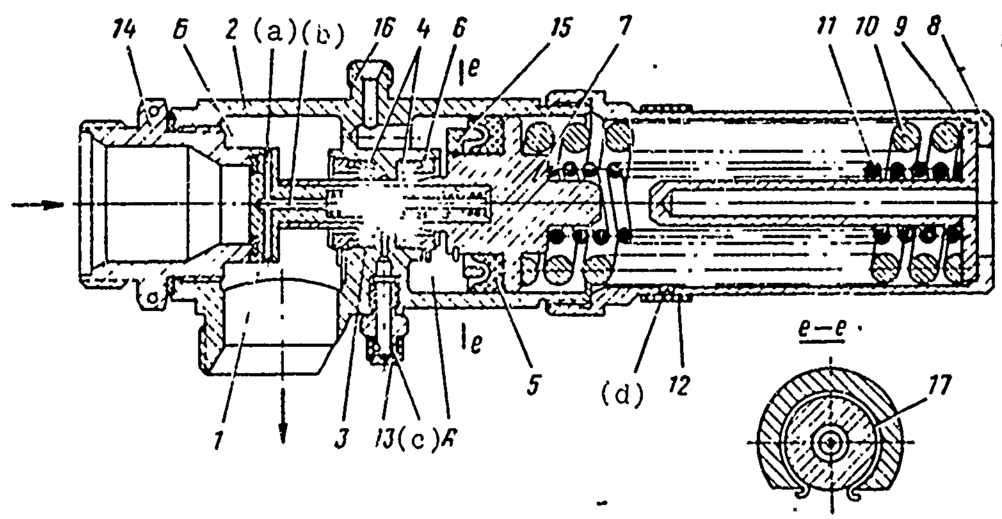

Fig. 2.7. A normally closed valve: A - controlling cavity; B - liquid cavity; a, b - bores; c, d - vent openings; 1 - valve; 2 - housing; 3 - spring; 4, 5 - seal rings; 6 - ring; 7 - insert; 8 - spring casing; 9 - detent; 10, 11 - springs, 12 - dust-protection ring; 13, 14, 16 - tube connections; 15 - ring; 17 - wire.

With a nominal value of the propellant flow rate the drop on the valve significantly exceeds the force of spring 3 and therefore the valve is completely open, up to the detent in insert 7. It is impossible to make a rigid connection of valve 1 with insert 7, in view of the possibility of a sudden pressure increase behind the assembly, which may occur during unstable transient processes. But when there is a flexible coupling through spring 3 valve 1 is covered and will not permit propagation of the pressure wave. To close the valve, the pressure from cavity 6 must be released through connector tube 16.

The controlling and liquid cavities are separated by two rubber seal rings 4. Vent is accomplished between the seal rings through connector tube 13, from which air and propellant vapors can flow (in case of penetration through the seal rings). The cavity of the spring is separated from the controlling cavity by rubber seal ring 5. In the case of nonhermeticity of seal rings 4 or 5 the air is removed through openings c and d. Vent openings c and d are covered by dust proof rubber rings. The rubber seal of valve 1 is vulcanized. The fastening of the seal rings is accomplished by wire locks 17.

During the entire time of storage of the assembly springs 10 and 11 press valve 1 to the seat of connector tube 14. Due to the prolonged and considerable loading the rubber of the seal of valve 1 ages relatively quickly and loses its elasticity, as a result of which the degree of hermeticity of the seal may decrease, especially at low operating temperature.

A significant shortcoming of the given design is also the "adhesion" of the rubber of valve 1 to the seat under the prolonged action of the spring pressure. When controlling air is supplied only insert 7 moves forcibly, while valve 1 is open under the pressure of the propellant, which overcomes the force of spring 3.

The phenomenon of rubber "adhesion[1]," which has been observed during tests at low temperature (-40°C) after prolong storage (about a year) at a temperature of +50°C, leads to a rather significant and nonuniform retardation of the opening of the valve.

Due to the presence of the mentioned defects the valve design requires additional finishing - with respect to the selection of the optimum seat profile, valve seal material and so forth.

A normally-open pneumatic propellant valve of the balanced type is presented in Fig. 2.8. Unloading is achieved by the use of duel-disk valve 2.

Figure 2.8 depicts the valve in the open position, without pressure in the liquid and controlling cavities. To put the assembly into the "open" position it is sufficient to supply controlling pressure to cavity A through connector tube 7; then valve 2 is raised, overcoming the force of spring 9.

To calculate the force from the compressed air, which keeps the valve in the closed position, it is assumed that the effective area $F_э$, which receives the air pressure, is equal to the area of the circle formed by the average diameter D_{cp} of bellows 4.

When propellant is supplied to the assembly inlet, it passes through opening a in valve 2 into cavity Б. Thus, the propellant pressure acts (on the side of the opening) on the area of the circle according to the average diameter of the bellows, with a deduction for the area of the ring formed by the diameters of both seats in the housing:

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[1]The phenomenon of rubber "adhesion" is examined in more detail in section 8.1.

$$F_{\tau}=\frac{\pi D_{cp}^{2}}{4}-\frac{\pi\left(D_{1}^{2}-D_{2}^{2}\right)}{4};$$

$$F_{\tau}=\frac{\pi}{4}\left(D_{cp}^{2}+D_{2}^{2}-D_{1}^{2}\right),$$

where F_T is the calculated area, which takes the pressure of the propellant;

D_1 is the diameter of the larger seat;

D_2 is the diameter of the smaller seat.

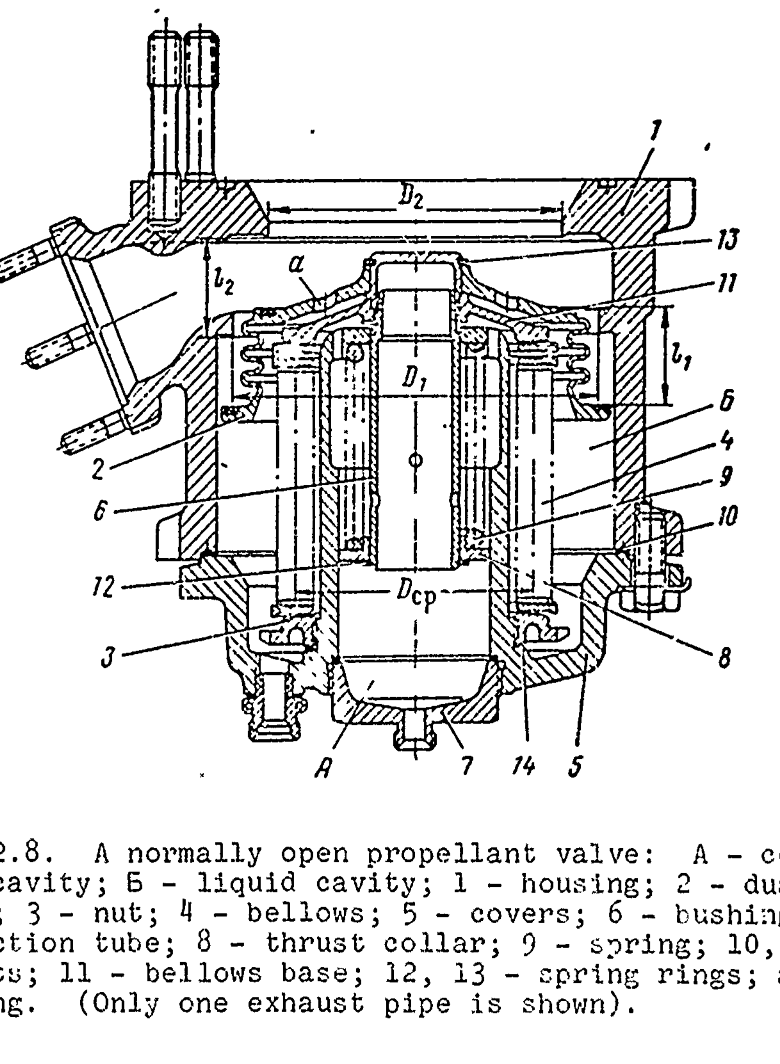

Fig. 2.8. A normally open propellant valve: A - controlling cavity; Б - liquid cavity; 1 - housing; 2 - dual-disk valve; 3 - nut; 4 - bellows; 5 - covers; 6 - bushing; 7 - connection tube; 8 - thrust collar; 9 - spring; 10, 14 - gaskets; 11 - bellows base; 12, 13 - spring rings; a - opening. (Only one exhaust pipe is shown).

It is easy to see that this area F_T is significantly less than area $F_c = \frac{\pi D_1^2}{4}$, on which the propellant pressure would act in the case where a known discharge system is used. Therefore the valve, depicted in Fig. 2.8, has a smaller size than a valve of the nonbalanced type would have, and may operate at a lower pressure value of the controlling air.

When using a discharge controlling air supply system it is possible to completely shut off the flow through the assembly at nominal propellant pressure.

Attention should be drawn to the fact that, if the area of the ring formed by the diameters of the seats is greater than the effective area of the bellows, i.e., $(D^2_1 - D^2_2) > D^2_{cp}$, then during the discharge of the air pressure from cavity A the opening of the assembly (with the presence of pressure p_T in cavity Б) may not occur. With such a design for opening the valve it is essential that the propellant pressure p_T be less than a certain value p_0, wherein

$$p_0 = \frac{4Q_{np}}{\pi\left[(D_1^2 - D_2^2) - D_{cp}^2\right]},$$

where Q_{np} is the spring pressure in the compressed state (with the valve closed).

In this assembly the most interesting construction is that of dual-disk valve 2. For normal hermeticity of the assembly it is necessary to ensure the sealing of valve 2 simultaneously on both seats of the housing. This requires the precise maintenance of distance l_2 between the seats in the housing and a correspondingly strict tolerance for dimension l_1 on the valve (see Fig. 2.8).

The sealing material employed for the valve is a very corrosion-resistant teflon, which is sufficiently hard under conditions of engine operation at temperatures of -50°C.

In view of the low elasticity of the sealing material valve 2 itself was manufactured in the form of a rather open elastic bellows. It has been provided for, that dimension l_1 between the ends of the seals for the valve will always be equal to or less than (for a value up to 0.3 mm) dimension l_2 between the housing seats. Because of this during the closing of the valve at the first moment the sealing occurs along the seat with diameter D_1. However, under the effect of the compressed air force the valve bellows was deformed (extended) and sealing was achieved along the seat with diameter D_2. Both seats are of the open type, with a height of 2 ± 0.2 mm, a width of 2.5 mm, and a radius equal to 0.5 ± 0.1 mm at the top.

When using propellants which admit the possibility of using rubber, the need for an open valve construction disappears. As a result of the elasticity of the rubber seal with corresponding accuracy for the manufacture of parts it is possible to ensure the required hermeticity (even with checks by compressed air) on both seats and with a rigid (and not elastic) construction of valve 2.

In the assembly depicted in Fig. 2.8, the width of the wall of valve 2 (on the bellows) amounts to 1.8 mm. Bellows 4 is two-layered and made of steel; the thickness of each layer is 0.2 mm; it is similar in construction to the bellows described above, i.e., equipped with inner and outer rings, which provide strength and rigidity.

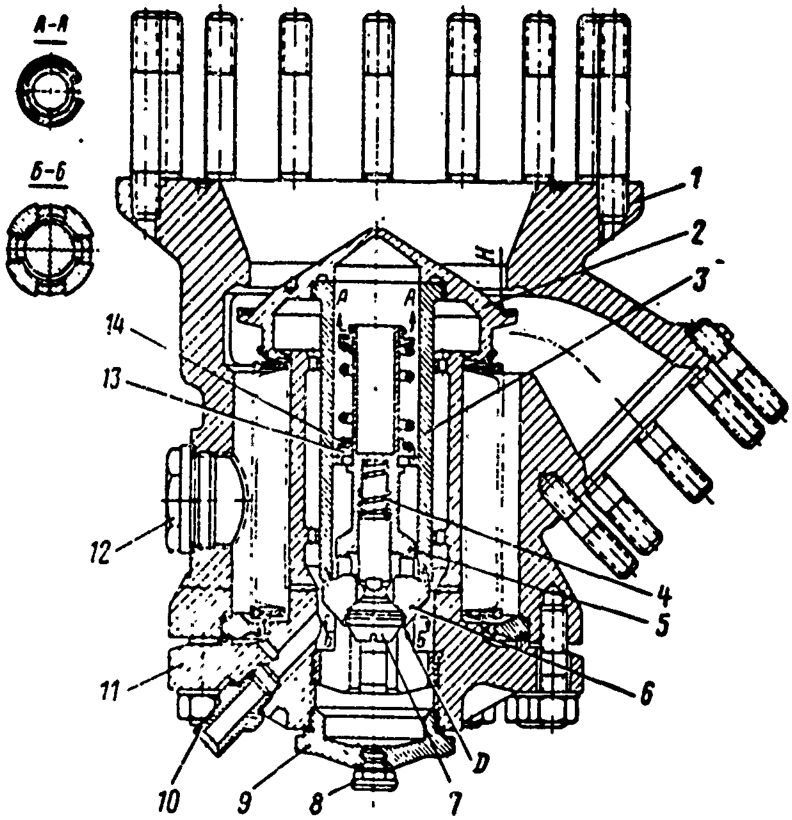

Fig. 2.9 Main oxidizer valve: 1 - housing; 2 - valve disk; 3 - sleeve; 4 - spring; 5 - support; 6 - cross-arm; 7 - explosive bolt; 8, 12 - plugs; 9, 11 - covers; 10 - connector tube; 13 - support; 14 - spring.

Bellows 4 is welded to base 11 and nut 3 by means of seam welding. The stationary ring of the bellows is fastened on a thread by nut three to cover 5. In base 11 is screwed sleeve 6, centered in cover 5. The force of spring 9, acting on sleeve 6, is absorbed by flat split snap ring 12; snap ring 13 holds valve 2 from axial movement relative to the bellows unit.

The sealing between housing 1 and cover 5 is ensured by thin aluminum gasket 10. To improve the hermeticity in the housing a groove has been provided, and in the cover - a tooth, which restrains the gasket.

The housing and cover are manufactured from cast aluminum alloy AL4.

The main oxidizer valve, shown in Fig. 2.9, handles liquid oxygen. It is a normally open pneumatic valve of the direct-action, non-balanced type (closing against the flow of the liquid).

The assembly has three fixed operating positions:

a) closed;

b) open for operation on prestage mode;

c) open for operation on main stage mode.

These valve positions are provided by a single locking mechanism.

When pressure is supplied to the controlling cavity through connector tube 10 disk 2 will be pressed against the seat of housing 1; the collar of sleeve 3, screwed into the disk of valve 2, will raise support 13 and squeeze spring 14.

Between sleeve 3 and cross arm 6, connected by bolt 7 with stationary support 5, a gap, equal in value to the valve's displacement, is formed. In this (closed) position liquid oxygen is supplied to the valve.

In order to open the valve for operation on prestage mode, the controlling pressure is released; the valve begins to open both under the action of the oxygen pressure at the inlet to the assembly, as well as under the effect of the pressure of spring 14. The movement of the valve (travel H) is limited by the detent of sleeve 3 to cross bar 6, which hangs on bolt 7. The magnitude of travel H is regulated by the degree of screwing in of bolt 7.

The opening of the valve for operation on main stage mode proceeds with the increase in oxygen pressure in front of the valve. With the gradual increase in pressure the position of the locking

mechanism does not change (the force is absorbed by bolt 7), then the bolt begins to be pulled out and at a given force (determined by the pressure drop on the disk) the bolt is torn out, as a result of which the valve opens abruptly for operation on main stage mode. The threaded portion of the bolt remains in stationary support 5.

A special feature of the valve design is the presence of an explosive bolt, which ensures the required opening of the valve for operation on main stage mode.

The amount of travel H, which limits the propellant flow rate in the preliminary stage, is determined by the water flow bench test of each individual valve sample proceeding from a given drop Δp at a certain flow rate of liquid. The amount of travel is recorded in the valve's log book.

To prevent the possibility of the bolt coming lose while the valve is in the closed position, spring 4 is inserted. To ensure stability of the process of transition to main stage mode it is necessary to achieve a constant value for the force of the rupture of the bolt. This is achieved by special means. Explosive bolt 7 is manufactured from various aluminum alloys or bronze, the tensile strength of which at temperatures above -130°C is practically temperature-independent[1].

The bolt has admittedly a weak spot - the neck, which is made by a special recessing tool to avoid stress concentration. The neck of the bolt is machine-finished to a smoothness of $\Delta 7$. The

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[1]The selection of material depends on the required strength of the bolt.

diameter of the neck of the bolt in the diagram is given only tentatively, and ultimately the diameter is determined (with an accuracy of up to 0.03 mm) separately for each rod, from which the bolts are manufactured, on the basis of test results for the rupture of samples of bolts, manufactured from the same rod. The mechanical qualities are preliminarily checked in every rod. After the bolts are manufactured, a significant number of them are selected for rupture testing with the aim of checking the manufacturing stability of the entire batch.

In case of necessity without special trouble the valve design permits the replacement of the bolt even on n assembled engine - for this it is only necessary to open cover 9 To replace the bolt it is necessary to ensure a previously existing value of travel H, which is achieved by the unchangeability of the position of the end face D (see Fig. 2.9), the distance of which from the face of cover 11 is easily measured.

In order to increase the stability of the process of valve opening for operation on prestage mode measures are taken to increase the air temperature in the controlling cavity (see section 8.2 for more details). For this, in plug 8 there is a discharge jet with a diameter of 0.5 mm. When the valve is in the closed position, air flows through the controlling cavity. This somewhat increases the temperature of the controlling cavity of the valve, which is cooled by the oxygen in the liquid cavity. Moreover, as a result of the flow of air the temperature of the explosive bolts is increased, which promotes the stability of its rupture.

The normally closed pneumatic valve, shown in Fig. 2.10, gives an example of an original solution to the problem of cutting down sizes. Instead of reducing the area on which the propellant pressure acts, here the area of action of the controlling pressure is artificially increased approximately twice. The assembly has

as it were two controlling cavities - A and Б, which are interconnected.

The assembly's moving system, shown in Fig. 2.10, consists of rod 2, on which by means of a wire stop ring the disk of valve 3 with bellows 4 welded to it is fastened; to this same rod 2 is fastened on threads piston 18 with seal ring 19 and support 5. The position of piston 19 relative to rod 2 is fixed with the aid of a stop washer and nut 14. The stationary ring of the bellows is welded to ring 15, compressed between housing 1 and guide 23.

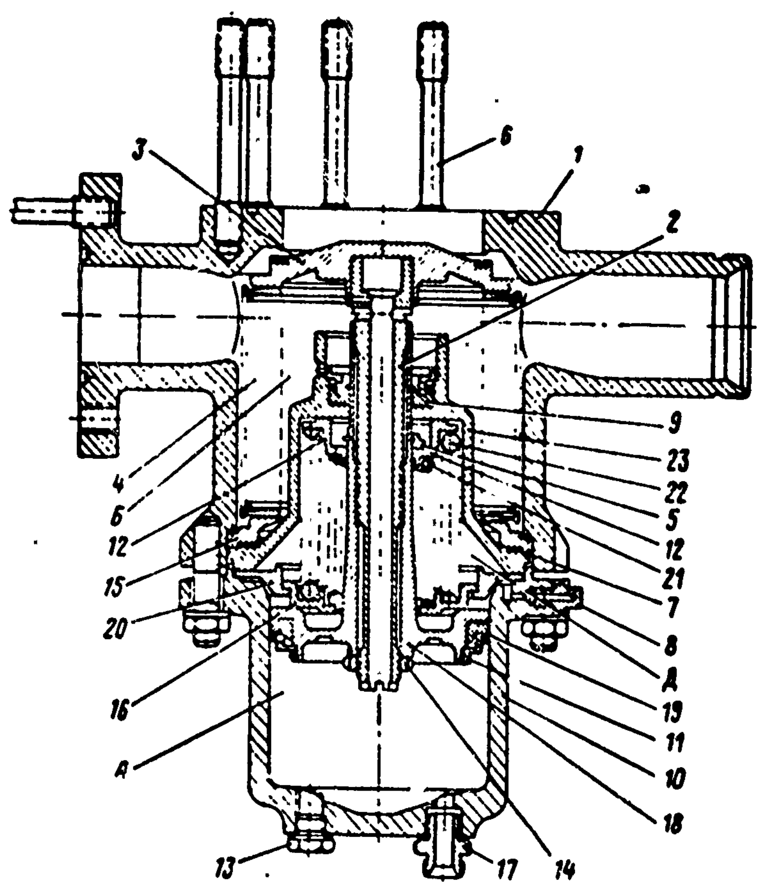

Fig. 2.10. A normally open pneumatic valve: A, Б - controlling cavities; Д - vent cavity; 1 - housing; 2 - rod; 3 - valve disk; 4 - bellows; 5 - support; 6 - pin; 7 - gasket; 8 - vent connector tube; 9 - seal ring; 10 - cover; 11, 15 - rings; 12 - bushing; 13 - plug; 14 - nut; 16 - spring retainer; 17 - connector tube; 18 - piston; 19 - seal ring; 20 - support; 21, 22 - springs; 23 - guide.

The operating scheme of the assembly is shown in Fig. 2.11. When controlling pressure is supplied to connector tube 17 (see Fig. 2.10) the valve is closed. The pressure of the controlling air, which is in cavity A - under the seal ring - and in cavity Б - inside the bellows, compresses the valve to the seat (see Fig. 2.11a). Clearance g is now formed between support 20 (see Fig. 2.10) and disk 15 (because of the shifting of the latter). Precisely the same clearance is formed between stationary bushing 12 and support 5.

There is no air between cavities A and Б (in cavity Д of springs 21 and 22), since seal ring 19 prevents air from getting into cavity Д from cavity A; seal ring 9 ensures the insulation of cavity Д from cavity Б. In the event of nonhermeticity of seal rings 9 and 19 or of gaskets 7 air, entering cavity Д, will be removed through vent connector tube 8, situated in cover 10, to the atmosphere (the vent holes in the connector tube are covered with dustproof rubber.

Force R from pressure p_y of the controlling air will equal:

$$R = \frac{\pi}{4} p_y \left(D_м^2 + D_{ср}^2 - d_м^2\right),$$

where $D_м$ is the outer diameter of movable seal ring 19 (after installation);

$D_{ср}$ is the average (effective) diameter of bellows 4;

$d_м$ is the inner diameter of stationary seal ring 9 (after installation).

Force R overcomes the force of the preliminary compression of springs 21 and 22, moves the retainer of spring 16 and expands bellows 4 (see Fig. 2.10).

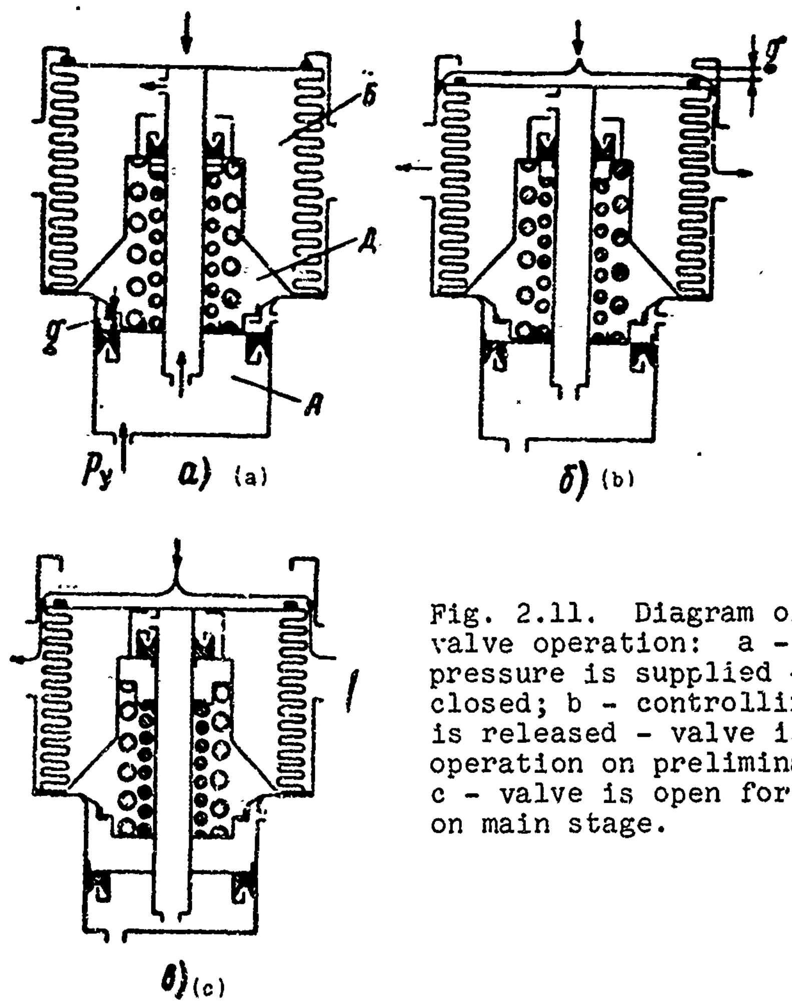

Fig. 2.11. Diagram of propellant valve operation: a - controlling pressure is supplied - valve is closed; b - controlling pressure is released - valve is open for operation on preliminary stage; c - valve is open for operation on main stage.

With the discharging of the controlling pressure springs 21 and 22 return the moving system to the initial position. Now clearance g is formed between disk 3 of the valve and the seat of housing 1. In this position the valve is open for operation on prestage mode (see Fig. 2.11b). The magnitude of clearance g determines the propellant flow rate in prestage mode. Clearance g is regulated by the degree to which rod 2 is screwed into piston 18. The required travel magnitude (clearance) is determined using hydraulic tests - by a bench flow test of the assembly.

With the buildup in the liquid cavity of the propellant pressure, acting on the bellows from outside, the valve is

opened for operation on main stage mode. With the valve completely opened disk 3 rests on guide 23. This position is shown in Fig. 2.11c.

When pressure is supplied to the controlling cavity (even when there is working pressure in the liquid cavity) the valve is closed (see Fig. 2.11a), in spite of the fact that the controlling pressure is substantially less than the propellant pressure in front of the assembly.

Thus, when pressure is supplied to the controlling cavity, disk 16 and the lower ends of springs 21 and 22 move; when pressure is supplied to the liquid cavity bushing 12 and the upper ends of these springs move. In both cases the compression of the springs is increased. Minimum compression takes place during operation on prestage mode (see Fig. 2.11b). The force of the springs under minimum compression is called the "force of preliminary compression" (this may be rather great). An increase in the spring pressure in the closed (see Fig. 2.11a) and open (see Fig. 2.11c) positions of the assembly is determined by the spring force. It is desirable that the spring force (the change in pressure per unit of length of compression) be minimal.

Many main propellant valves are made in such a way, that when there is no pressure in the liquid and controlling cavities they are open for operation on prestage mode, i.e., open only for part of their possible travel; they are closed by supplying controlling pressure, and by releasing it and by supplying pressure to the liquid cavity they are completely opened. Therefore the diagram presented in Fig. 2.11 is just as typical for propellant valves as the diagrams shown in Figs. 1.3, 1.4, and 1.5.

The manufacture of the assembly does not present any special difficulties. The bellows is similar to the bellows shown in Fig. 2.2. To ensure better hermeticity, at the ends of ring 15

(see Fig. 2.10), housing 1 and guide 23 annular grooves are made, into which the gasket material flows. The grooves are made in such a way that their diameters do not coincide in mated parts. The gaskets are made of copper and are thin (thickness of 0.5 mm).

A <u>normally open starting check valve</u> of the balance type is shown in Fig. 2.12. This is a bench unit, which is used on test stands for testing an engine as a whole or individual units of it.

To close the assembly compressed air is supplied to connector 14 through upper control cavity A. The compressed air, acting on piston 4, compresses spring 8 and expands bellows 7. Now small valve 2 first seats its valve in its seat, which is made in basic valve 1, and then valve 1 settles into its seat, which is made in housing 6.

When the propellant, which is under high pressure, is supplied to the inlet of the assembly (to flange b), the force of the pressure of valve 1 against its seat is increased. The force pressing against valve 2 is also altered somewhat. The magnitude of change of the force pressing valve 2 against its seat, which is made in valve 1, is determined by the area, which is the difference of the areas of the seat of valve 2 and bellows 7, on which the propellant pressure acts. Since the area of the seat of valve 2 is somewhat greater than the area of the average diameter of bellows 7, the force applied to valve 2 also increases when pressure is supplied.

In order to open the starting valve, the controlling pressure is released from cavity A. Then spring 8, overcoming the propellant pressure in the inlet cavity of the assembly, raises valve 2 from its seat. When valve 2 lifts off its seat, the pressure inside cavity B falls sharply, since propellant enters the output main line through openings A and through the clearance between the

seal of valve 2 and the seat made in valve 1. The flow of propellant through the diametric clearance between housing 6 and valve 1 (the clearance is approximately 0.1-0.5 mm) does not ensure the maintenance of the pressure in cavity B, since the area of this clearance is clearly significantly less than the area of openings A.

As soon as the pressure in cavity B falls, valve 1 begins to rise together with valve 2 under the effect of the propellant pressure on the area of the ring, formed by the outer diameter of valve 1 and the diameter of the seat, built into housing 6. The amount of travel of valve 2 is limited by the detent of piston 4 to the end of cover 9.

The propellant pressure acting on the area of bellows 7 (with respect to the average diameter) assists the spring in maintaining the valve in the open position. With pilot valve 2 fully open, main valve 1 must hang on valve 2, resting on washer 5 with spring stop ring 3. Stop ring 3, placed in the bushing of valve 1, guarantees the raising of valve 1 in case it gets jammed in housing 6 (or in case of other abnormalities in the valve's operation). When the assembly is in the open position, the propellant pressure in cavity B is equal to the pressure under valve 1 or, more precisely, somewhat higher than it[1].

To close the valves it is sufficient to supply compressed air to controlling cavity A. When the air pressure overcomes the force of spring 8 and the force of the propellant pressure on the area of bellows 7, valve 2 starts to close. With the downward movement of pilot valve 2 main valve 1 also starts to move. Both the pressure drop (at the first moment) and the direction of the propellant flow facilitate this. If the movement of valve 1 is retarded (for any reason) pilot valve 2 will close it forcibly.

[1]For the value of hydraulic loses.

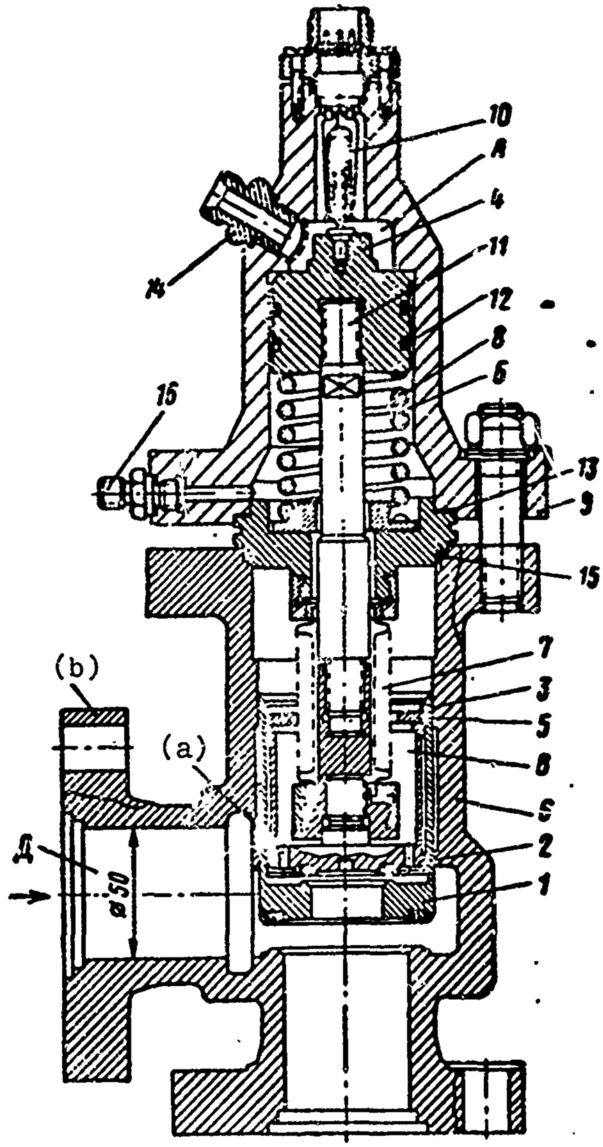

Fig. 2.12. A normally open balanced valve: A - controlling cavity; Б - auxiliary controlling cavity; В - propellant cavity; Д - inlet pipe; 1 - main valve; 2 - small valve; 3 - stop ring; 4 - pistor; 5 - washer; 6 - housing; 7 - bellows; 8 - spring; 9 - cover; 10 - signal indicator; 11 - rod; 12 - seal ring; 13, 15 - gaskets; 14, 16 - connector tubes; a - bore in valve 2; b - inlet flange of housing 6.

The closing of the starting valve occurs sharply. To decrease the rate of closing we should, if possible, reduce the diametric clearances between valve 1 and housing 6 or insert a labyrinth seal between housing 6 and valve 1 - in order to restrict the supplying of additional propellant to cavity B.

To ensure failsafe functioning of the valve cavity Б is also frequently employed: with the discharge of pressure from cavity A (with the opening of the valve) the controlling pressure is supplied through connector tube 16 to cavity Б[1]. And on the other hand, while closing, when pressure is supplied to cavity A, the pressure is discharged from cavity Б[2]. The pressure supply to cavity Б during opening significantly raises the force which opens the valve.

Let us note that the diameter of the connection tube of cavity Б is significantly less than the diameter of the connection tube of cavity A, although the volume of cavity Б itself is substantially greater than the volume of cavity A. Because of this, in view of the relative slow drop in pressure in cavity Б, a retardation is achieved during the closing of the valve.

Because of the fact that the valve is an assembly of the balanced type (as a result of the utilization of pilot valve 2), it was possible to use a relatively weak spring 8 (a working pressure of 125 kgf) and a small piston diameter with a comparatively low controlling pressure in cavity A (not exceding 100 at), although the valve may operate at a liquid pressure of up to 300 at.

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[1]The controlling pressure in cavity Б is small - approximately 20-50 at. This is done in order to improve the operating conditions of bellows 7, since with the presence of additional pressure bellows 7 will break down earlier, cracks will form on it and its service life will be reduced approximately twice in comparisor with the service life, which it would have if operating without pressure in cavity Б.

[2]Such valves are called dual-control valves.

The assembly has electric signal indicator 10 to ensure the possibility of recording the movement of the moving system. If necessary, instead of the signal indicator a travel sensor can easily be installed to record on an oscillograph the processes of valve movement.

Teflon was used as the seals for valves 1 and 2. However, the use of teflon for valve 1 sharply reduced the permissible operating life as a result of the breakdown of the seal.

For a test bench assembly the value of the service life is extremely important, therefore instead of the fluoroplastic for valve 1 aluminum, pressed under great pressure into the seat, which had a dovetail shape, was used.

The use of a soft aluminum seal on the valve and a hard seat (the housing is made from steel) provides acceptable hermeticity (with water or propellant) during the short-term effects of high pressures.

To separate control cavities A and Б rubber rings 12 are used. The separation of air cavity Б and liquid cavity B is provided by the bellows. A series 6-layer bellows, manufactured from steel 1Kh18N9T, is used for this. For normal operation of the bellows with the above-cited parameters compression of the bellows greater than 0.8 mm for one corrugation should not be permitted.

The described assembly can be used for handling various fuels and oxidizers at temperatures within the range of ± 50°C.

When using low-boiling oxidizers the reworking of the valve's design was required. It was necessary to separate and thermal-insulate controlling cavities A and Б from the liquid cavity (this was achieved with the aid of inserts made from heat-insulated material) in order to alleviate the work of rings 12,

which separate the controlling cavities, and to examine the clearances in the movable joints. The fluoroplastic seal in valve 2 proved to be short-lived under these conditions, and a "metal-to-metal" seal was employed. The altered valve design is shown in Fig. 2.13.

Figures 2.12 and 2.13 show the assemblies without pressure in the liquid and controlling cavities.

The <u>repeating propellant valve</u>, shown in Fig. 2.14, is, in contrast to the earlier examined assemblies, not a pneumatic valve, but an electric valve (EV) of the discharge type, controlled by an electromagnet. It is designed for main lines with a low propellant flow rate.

With the assembly in the "closed" position (the electromagnet is de-energized) the propellant is in cavity B, from which it flows through diametric clearance δ between valve 5 and housing 7 (clearance δ is 0.1-0.14 mm; the valve diameter is 17 mm).

When the propellant is under pressure, then main valve 5 is pressed against the seat of housing 7 by this pressure and by the force of spring 8, while pilot valve 4 presses against main valve 5.

When voltage is applied to the terminals of the electromagnet of armature 1, tightened to yoke 2, it moves downward. The pressure of the armature is transmitted to rod 3, which, in turn, presses against pilot valve 4.

Valve 4 can be lowered relative to main valve 5, overcoming the spring force and the propellant pressure, to a value of ~0.5 mm up to the detent in stop rings 6. Further movement of valve 4 is possible only together with valve 5. However, the thrust force of the magnet would be insufficient to move valve 5, if the

pressure in cavity B remained unchanged. But as soon as valve 4 moves somewhat from its seat, the pressure beneath valve 5 falls abruptly (on valve 4 along the entire length there are flat keyways, along which the propellant goes to the central opening in valve 5); the entrance of propellant through diametric opening δ cannot support pressure in cavity B, therefore the pressure which exists above valve 5 moves it down.

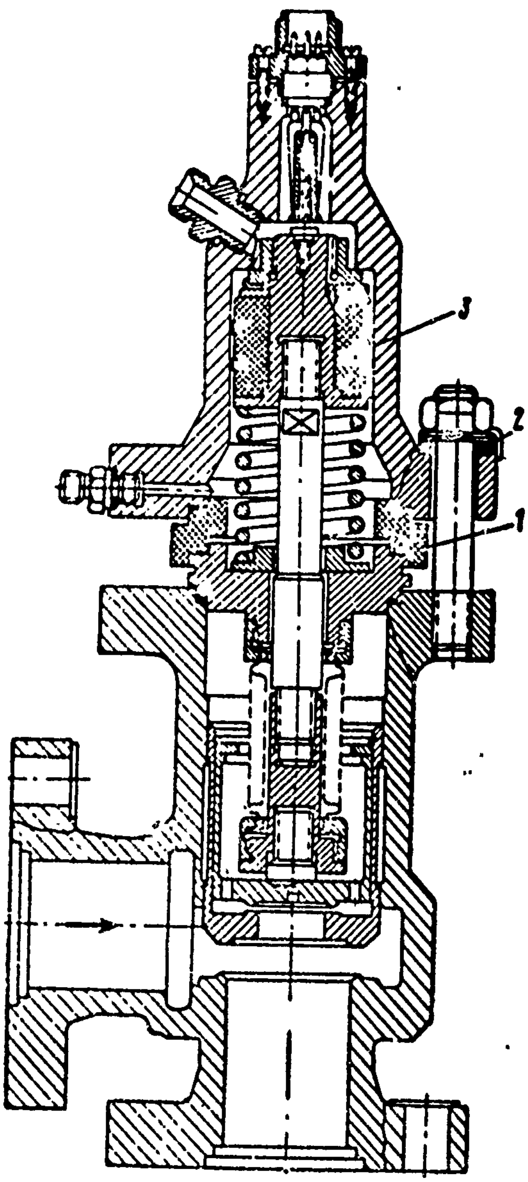

Fig. 2.13. A normally open valve designed for handling low-boiling liquids; 1 - adapter; 2 - ring; 3 - bushing.

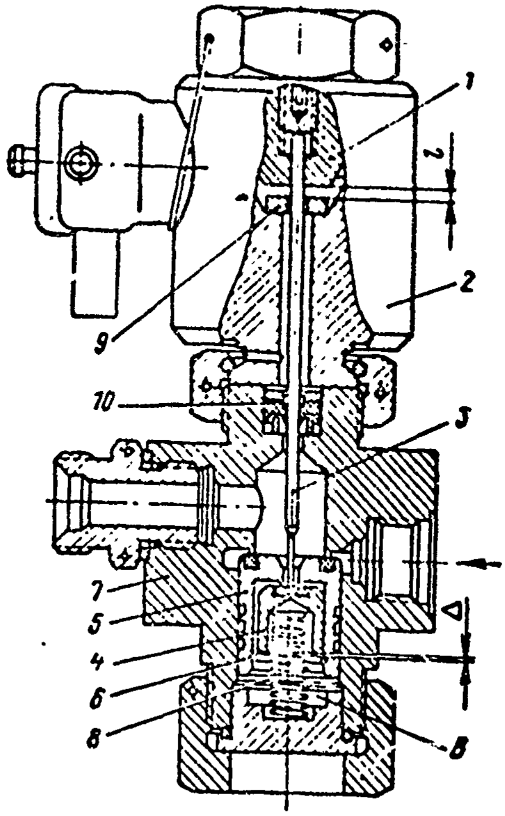

Fig. 2.14. A release-type electrically operated valve: B - cavity; 1 - armature; 2 - yoke; 3 - rod; 4 - pilot valve; 5 - main valve; 6 - stop ring; 7 - housing; 8 - spring; 9 - adapter; 10 - seal ring.

The presence of a labyrinth in the outer diameter of valve 5 facilitates the precise opening of the EV - increases the hydraulic resistance of clearance δ, i.e., prevents the overflowing of the propellant into cavity B.

The moving of the system of valves 5 and 4 will proceed up to the moment of detent of armature 1 in copper insert 9 - to the value l of travel of the armature. The travel of valve 4 is always equal to the travel of the armature of the electromagnet. The position of main valve 5 in the open position of the assembly under steady state conditions is determined by the relationship of the weight of the valve and the pressure drop on it. The pressure above valve 5 is somewhat less than the pressure below the valve because of the hydraulic resistance during the flow of propellant through the seat of the housing. The pressure

drop on valve 5 depends on the propellant flow rate through the assembly, the travel of the armature and the magnitude of the diametric clearance δ.

As a rule, the pressure difference on both sides of valve 5 exceeds its weight and consequently valve 5 is slightly raised above valve 4. In certain cases during oscillations of the flow rate valve 5 may vibrate.

To open the valve the electromagnet is de-energized. Then by the force of spring 8 and the increase (during the seating of valve 4 on the seat built into valve 5) pressure drop valves 4 and 5 are raised upward - the EV is closed.

The value of the friction in the seal ring has a significant effect on the operation of the assembly. An increase in friction can lead to failure or the retardation of closing of the valve.

During assembly strict control of the straightness of rod 3 is essential, since bending in it can lead to jamming in the moving system.

2.2. ONE-TIME VALVES

The most popular one-time valves are explosive valves, i.e valves controlled by explosive cartridges.

A normally open direct-action explosive valve, with venting of the solid-reactant gases, is depicted in Fig. 2.15. It is designed for cutting off the flow of liquid oxygen.

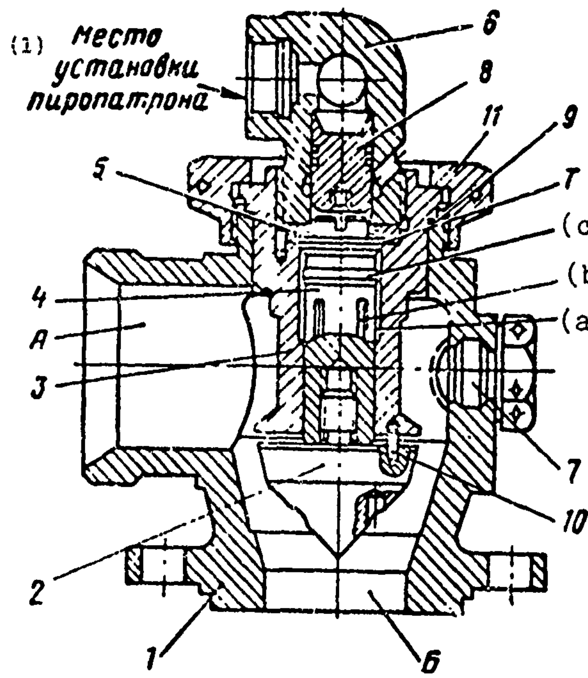

Fig. 2.15. A normally open direct-action explosive valve with vent: A - inlet cavity; Б - outlet cavity; T - collar (bead) of rod; 1 - housing; 2 - valve; 3 - guide; 4 - rod; 5 - ring; 6 - corner iron; 7 - plug; 8 - piston; 9 - separator; 10 - pin; 11 - coupling nut; a - annular opening; b - groove; c - neck of rod (the location of the second explosive cartridge is not shown).
KEY: (1) Explosive cartridge installation site.

Up to the moment of triggering of the explosive cartridges the propellant freely enters from inlet cavity A into outlet cavity Б, flowing around valve 2. With the triggering of the explosive cartridges (the explosive cartridges are not shown in Fig. 2.15) the solid-reactant gases act upon piston 8. Piston 8 moving, presses against rod 4, shearing off the collar T which is pressed by corner iron 6 with the aid of ring 5, and which holds rod 4. After the shearing off of the collar of rod 4 valve 2 under the action of piston 8 and the flow of liquid oxygen seats itself on the conical surface of housing 1; in this position the valve is closed, and outlet cavity Б is separated from inlet cavity A. Valve 2 is tightly wedged in housing 1. The force exerted on valve 2 from the housing amounts to 2000-3000 kgf.

The valve triggering time (from the moment of supplying a voltage to the explosive cartridge to the cessation of the flow) amounts to about 0.02 sec.

The nonhermeticity of the sealing of valve 2 in the seat of housing 1 (after triggering) with a liquid oxygen pressure of 80 at is not great - leakage does not exceed 15 cm^3/sec of gaseous oxygen (reduced to atmospheric pressure and a temperature of 10-15°C).

The separation of the solid-reactant gases from the liquid oxygen is accomplished by piston 8. In the lower part (up to the labyrinth grooves) the piston sits in corner iron 6 with a tight fit along the cylindrical surface (a second-class press fit). The assemblage of the piston with the corner iron is done with a force of 0.5 kgf with the cooling of the piston to -20-30°C and heating of the corner iron up to 200°C. Under working conditions the interference of the piston is increased relative to the interference existing at normal temperature, as a result of the cooling of the assembly by the flowing liquid oxygen.

The hermeticity of the oxygen cavity up to triggering of the explosive cartridges is ensured by a thin (0.4 mm) copper cadmium plated separator 9 (by adjustable coupling nut 11) and bead (collar) T of rod 4 (the width of the bead during manufacture equals 0.7 mm, compression during assembly amounts to 0.3 mm). The relative position of corner iron 6 and guide 3, which ensures the hermeticity of the bead, is fixed by cement.

With the triggering of the explosive cartridges the grooves made in piston 8 promote the reduction in penetration of the solid-reactant gases along the diametric clearance between the piston and the corner iron. After the piston moves 10-12 mm, the vent openings in the corner iron open, and the solid-reactant gases exit into the atmosphere.

With the movement of rod 4 the liquid oxygen, compressed in radial clearance a between the rod and guide 3, along grooves b in the rod emerge into cavity A - thereby preventing the possibility

of a pressure increase in clearance a, which might impede the movement of rod 4.

After the shearing of the bead of the rod the seal of the oxygen cavity is ensured by press fit of rod 4 in the guide.

The axial and diametric dimensions of the parts are selected so as to almost completely eliminate both the leakage of oxygen into the vent openings of the corner iron, and the entrapment of oxygen in radial clearance a.

The threaded connection of valve 2 with rod 4 is fastened with the aid of pin 10, pressed into valve 2 and seated in the seat of guide 3 (during assemblage rod 4 is screwed into valve 2, which is held by a special wrench and by pin 10). The screwing together of parts 2 and 4 is accomplished without using any lubricant so as to avoid an explosion during the operation of the assembly. To reduce friction one can use a lubricant-impregnated chamois to rub the threads of housing 1, nut 11 and separator 9. Excessive application of lubricant is not permitted - again, due to the danger of explosion when handling liquid oxygen.

With the same purpose - to minimize the danger of explosion - it is essential to pay especial attention to the cleanness of the parts. All parts are thoroughly degreased prior to assembly. The presence of traces of lubricant or any impurities in the internal cavity of the assembly (especially on the surfaces of adhesion of valve 2 to housing 1 and on the surfaces of the seat of rod 4 in guide 3) can during pronounced impact lead to an explosion.

Housing 1 and valve 2 are manufactured from aluminum alloy EV. Rod 4 is made from soft aluminum of brand AD-M; the material of guide 3 and of ring 5 is stainless steel of brand 1Kh18N9T; the material of the piston is heat-resistant steel, and the material of the corner iron is alloy AK6.

To ensure reliable operation of the assembly the installation of two explosive cartridges is provided for - in case one of the explosive cartridges fails to operate due to a breakdown in the continuity of the electrical circuit. Therefore, a check of the operation of the explosive valve (the precision of operation and the hermeticity of all the seals after triggering) is accomplished both with one explosive cartridge and with two of them.

The magnitude of the clearances and the interferences of the parts are determined from the conditions of operation of the valve under deep cooling. During testing under normal conditions such an assembly (water is used instead of liquid oxygen) does not guarantee a reliable wedging of valve 2 in the housing. As a result of the increased friction during the movement of rod 4 in guide 3 (because of the greater interference in comparison with that existing at low temperature) valve 2 cannot reach its seat.

A sharp impact during valve triggering, the heating of parts during the movement of the rod, during the breaking of the bead, accidental contamination, or finally, the breakthrough of solid-reactant gases - all this creates a background for a possible explosion in the liquid oxygen medium. In view of this, it is desirable to replace the aluminum alloys with alloys of copper (bronze, brass). Copper alloys are more heat-conducting, i.e., they more quickly remove heat from the heating center, and they have a higher ignition temperature.

The direction of flow of the solid-reactant gases is of basic significance for the reliability of the operation of the valve. With the arrangement of the explosive cartridge along the axis of movement of the isolating valve it is possible to use an explosive charge with a lesser weight of the explosive composition than with the positioning of the explosive cartridge perpendicular to the direction of movement of the valve, as is shown in the

assembly depicted in Fig. 2.15. The presence of hydraulic loses during the reversal of the flow of solid-reactant gases substantially decreases the impulse affecting the shearing of the bead.

A normally-open direct-action explosive valve, shown in Fig. 2.16, is one of the simplest designs of such type. The valve serves to cut off the flow of propellant; the operating temperatures are ±50°C.

Before the explosive cartridge is triggered, the propellant flows without difficulty around valve 1, entering the main discharge lines. With the bursting of the explosive cartridges[1] by the pressure of the solid-reactant gases neck 6 in valve 1 is broken and the valve is wedged in housing 2.

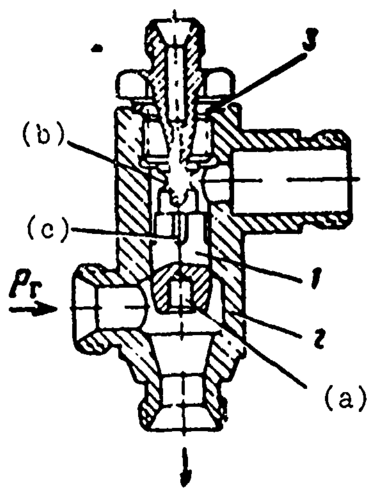

Fig. 2.16. A normally opened direct-action explosive valve: 1 - valve; 2 - housing; 3 - separator; a - bore; b - valve neck; c - milling (groove).

The assembly consists of only a few parts. Valve 1 is screwed on threads into housing 2. The hermeticity of the joint is ensured by seal 3.

[1]The explosive cartridges are not shown in Fig. 2.16.

Such a design of an explosive valve is used for operation only with those types of propellant, which do not explode in coming in contact with solid-reactant gases under high pressure, formed during the combustion of the explosive cartridges. The solid-reactant gases are removed through the bore in the part of the valve which remains stationary.

A vent tube is connected to the thread. This vent tube, as a result of the high values of temperature and the speed of the solid-reactant gases, burned away systematically, especially frequently at the point of bending - near the explosive valve. Therefore, instead of a tube it is better to connect a closed volume to the connector tube.

To increase the reliability of the wedging (to decrease the rigidity of valve 1) bore a is inserted in the body of the valve. On the surface of the valve additional shallow millings are also made - grooves c. The purpose of these grooves is to soften the hydraulic shock by means of connecting the propellant cavity with the drainage cavity through the bore in the body of the valve during closing of the valve (and after closing). The entry of the oxidizer into the solid-reactant gas medium brought about their afterburning and caused an increase in temperature, which promoted a hot spot on the vent tube.

The housing and valve are made from aluminum alloy of brands EV. The seating of the valve in the housing (along the cylindrical surface) is a medium, third-class (plain) fit. The seat in the housing is made with a conicity angle of 15°; the valve is rounded. Both the seat and the valve are machined to a highly polished finish.

With the valves set up on the test bench with a long hydraulic main line behind it (Fig. 2.17a) rebounding of the valve off its

seat was observed. This can be explained by the abrupt pressure increase in the output conduit during rapid movement of the valve. Recoil was observed with particular frequency, when buffer chamber 3 had a small size and was completely filled with liquid. With the creation of a gas phase in buffer chamber 3 (an air cushion of a certain pressure[1] was specially left behind) cases of rebounding were more infrequent. With the installation of an explosive valve with short tube 2 according to the scheme shown in Fig. 2.17b the working conditions of the explosive valve were even more improved, since the volume of gas in chamber 3 better dampened the pressure increase.

The installation of such valves is recommended in the direct proximity to the combustion chamber or gas generator, so that the hydraulic resistance between the explosive valve and the gas phase will be small.

With respect to preventing the occurrence of a high pressure peak at the valve inlet during triggering and to increasing the reliability of wedging a modification of the explosive valve shown in Fig. 2.18 has been more successful.

At the inlet cavity of this valve there is set up diaphragm 4, connected either with vent tubing or with the damping volume (Fig. 2.18 shows the connector tube, which can be either stoppered or butt-jointed with the vent tube). With an increase in pressure the diaphragm is destroyed (the pressure of destruction of the diaphragm is especially worked out) and this stops the pressure increase, since the auxiliary volume is connected.

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[1]The pressure in chamber 3 is created by gas pressure reducer 9 with the opening of the EPV (8). After this, the flow of liquid is ensured by the opening of EV (4), and then by EV (5). With high volumes of chamber 3 tube 11 and EV (5) cannot be used.

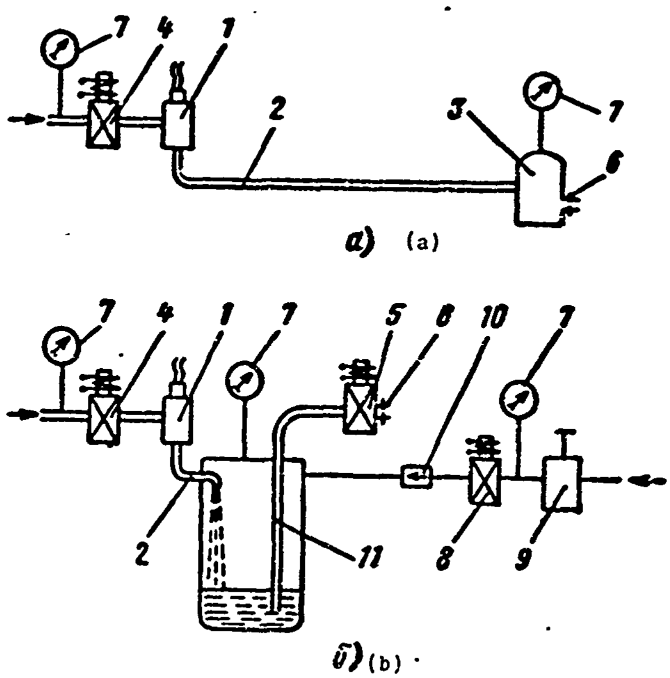

Fig. 2.17. Diagram of the set up for testing an explosive valve: a - difficult conditions for triggering the explosive valve (recoil of the valve is possible); b - improved operating conditions (the possibility of recoil of the valve is reduced); 1 - explosive valve; 2 - connection conduit; 3 - buffer chamber; 4, 5 - electrically operated valves; 6 - flow-rate nut; 7 - manometer; 8 - electro-pneumatic valve; 9 - gas pressure reducer; 10 - check valve; 11 - outflow conduit.

The design of valve 1 is also somewhat changed. The solid-reactant gases are not removed after triggering - the upper part of the valve is a plug. Vent of the solid-reactant gases is eliminated[1].

A direct-action normally closed explosive valve with vent of gases is depicted in Fig. 2.19.

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[1]The valve pressure force after triggering for the assembly depicted in Fig. 2.6 did not exceed 500 kgf, while after modification it reached as high as 4000 kgf.

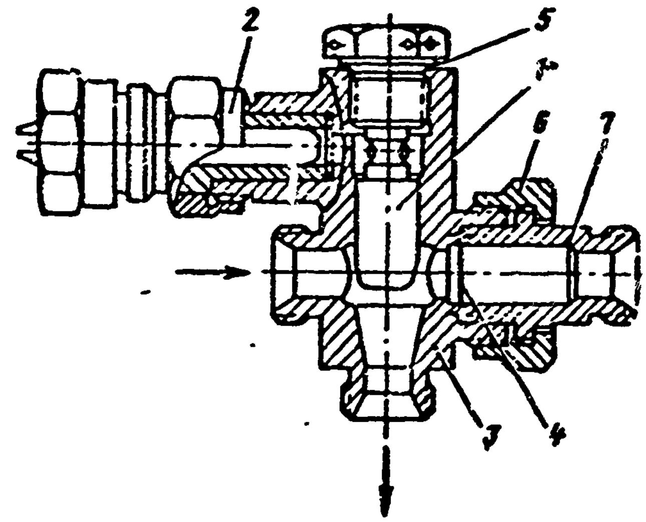

Fig. 2.18. Direct-action explosive valve, shown in Fig. 2.16, after modification: 1 - valve; 2 - explosive cartridge; 3 - housing; 4 - diaphragm; 5 - seal; 6 - nut; 7 - connector tube.

Such an assembly can be used as a servicing, vent and starting valve with low pressures and flow rates and can be operated within a temperature range ±50°C.

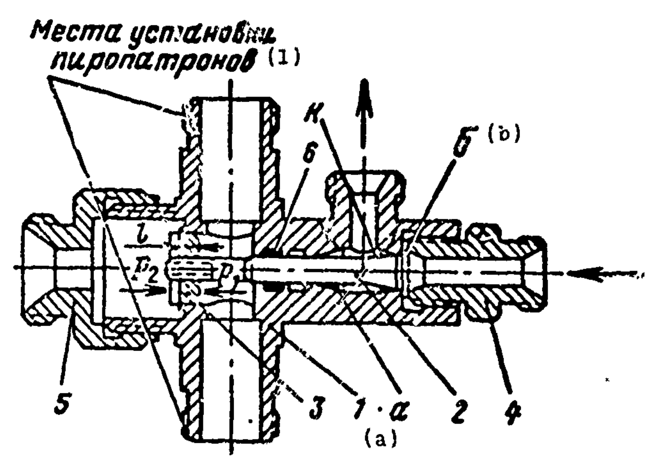

Fig. 2.19. A normally closed direct-action explosive valve with vent of solid-reactant gases: 1 - housing; 2 - valve; 3 - piston; 4 - connector tube; 5 - connector tube for venting of solid-reactant gases; 6 - rubber ring; a - bore hole; b - valve bead; K - valve cone.
KEY: (1) Location of explosive cartridges.

With the triggering of the explosive cartriges (the explosive cartridges are not shown in the diagram) the solid-reactant gases, acting on the area of piston 3, ensure the shearing of head b of valve 2, which is restrained by connector tube 4. During movement cone K of valve 2 is wedged in bore a of cone 1. The liquid or gas, conducted to connector tube 4, has free acess to the main line through the connector of the housing. The solid-reactant gases escape outside through connector tube 5.

The valve is manufactured from stainless steel, and the remaining metal parts - from aluminum alloy AV.

Piston 3 is screwed onto the rod of valve 2 up to the detent in housing 1 and is fastened in this position using cement. In screwing on the piston one should avoid the use of excessive force to prevent premature shearing of the valve collar.

The magnitude of the external diameter of piston 3 and the radial clearance between it and housing 1 determine the speed and force of wedging of the valve.

To ensure the hermeticity of the liquid cavity of the assembly during the movement of the valve there is rubber ring 6. It prevents the possibility of the penetration of the solid-reactant gases into the propellant. During operation with such propellants, which require a high degree of hermeticity, two rubber rings, installed serially one behind the other, are employed, and provide venting between the rings.

In the adjustment of such explosive valves for operation under various temperature conditions it is necessary to use care in selecting the weight of the moving parts, the charge of the explosive cartridges, and also to strictly control the tolerance values in the moving system of the valve.

In testing an explosive valve with a single explosive cartridge (in case one of two explosive cartridges fails) under low temperature conditions, when the strength of the collar is maximum, and the pressure developed by the explosive charge is minimal, it is necessary to limit the value of the clearance between piston 3 and housing 1. With an excessively great clearance the force of one explosive charge may be insufficient to shear the collar of valve 2.

However, during the operation of the explosive valve with two explosive cartridges under conditions of maximum working temperature and with minimum piston clearances the breaking of the stem of valve 2 is possible due to the emergence of large inertial forces at the moment the valve is wedged. Therefore, the mass of the piston must be minimal, i.e., the piston must be manufactured from light metal, for example, from aluminum. When using aluminum one should eliminate the possibility of a hot spot (from the action of the solid-reaction gases) of the metal on the threaded joint of piston 3 and valve 2; it is recommended that the threads be filled with cement ED-6, which simultaneously holds the joint, and seals the threads - it does not allow the hot gases to elute the aluminum.

Sometimes in such valve designs the rebounding of the valve from the valve seat may occur.

Right after the triggering of the valve (the breakoff of the collar and the shifting of the valve to the left) the solid-reactant gases, pushing toward connector tube 5, exert pressure on piston 3; the force of this pressure p_2 under certain conditions begins to exceed the force from pressure p_1. In this case the valve starts to move backwards and occupies its initial position - recoiling of the valve occurs.

During recoil (return) of the valve

$$p_2F > p_1(F - f),$$

where F is the area of piston 3;

f is the area of the stem of the valve 2.

The actuality of recoil can be established from the characteristic imprint from the seat on conical surface K of the valve.

Various elements of the design of an explosive valve have an effect on the phenomenon of recoil:

a) length l of the seat diameter of piston 3.

The greater is length l, the more abruptly will the pressure to the right of the piston increase, the longer the action time of this pressure and the more abruptly will the valve move to the left and the more deeply will it be wedged:

b) the diametric clearance between piston 3 and housing 1.

The greater the diametric clearance, the more quickly will the pressure increase to the right of the piston and the more probable will "recoil" of the valve be:

c) the flow-through cross section of the vent connector 5.

The smaller the flow-through cross section of connector 5, the more likely the probability of "recoil" of the valve: since the pressure drop on piston 3 is reduced. (As a result of modification of the design of an explosive valve the internal diameter of connector 5 was increased from 4 to 10 mm):

d) the charge of the explosive cartridges.

The lower the weight of the charge of the explosive cartridges, the more probable "recoil" is. With large charges, as a result of abrupt movement, the wedging goes deeper, and the valve cone is advanced further to the left. At the highest pressures developed by explosive cartridge (for example at elevated temperatures), with a certain combination of tolerances the "forcing through" of the valve through the housing even more deeply is possible, then is required. Therefore it is desirable to have a detent for piston 3;

e) the surface roughness of cone K.

For a greater reliability of wedging it is desirable to have not a smooth surface for the valve cone, but to make a row of annular scribes - "jags" on it. The cutting of this "jag" into the housing ensures the wedging of the valve. This method of "jag" is one of the efficient methods for combatting "recoil".

The presence of a vent for the solid-reactant gases (through connector tube 5) is in many cases undesirable from the viewpoint of the operation of the explosive valve: the triggering of the explosive cartridge releases through the vent hole a flow of hot gases and a sheaf of flame, which can start a fire. Therefore it is desirable to have a valve without a vent hole for the solid-reactant gases. However, as was shown above, the absence of a vent hole presents difficulties with respect to providing reliable wedging.

By using a "jag" on the housing, and having selected the necessary piston length and the clearance between the piston and housing, we succeeded in developing a design (Fig. 2.20), which ensured reliable operation of the explosive valve with venting into closed cavity A.

The construction of the explosive valve shown in Fig. 2.20 is basically similar to that depicted in Fig. 2.19. The isolation of the solid-reactant gases was achieved using stopper 7 and separator 8. The method of sealing the gas cavity was also changed. The separation of the gas and propellant cavities is provided for here by fluoroplastic bushing (seal ring) 6, which is seated with a certain interference on valve 2. The pressure of the solid-reactant gases presses the seal ring to the housing, increasing the hermeticity. The shape of detent 3 is somewhat changed. Detent 5, limiting the movement of the piston, is introduced.

The valves shown in Figs. 2.19 and 2.20, serve as prototypes for the construction of a check unit for explosive valves, installed on main propellant lines of large diameter.

An explosive pin for propellant valves is depicted in Fig. 2.21. The protruding end of the rod of pin 7 is located in shank 3 of the locking mechanism of the propellant valve and holds it in the given position. With the triggering of the explosive cartridge the gas pressure acting on piston 5 draws the pin from the shank, breaking bead b, which is made as one piece with pin 7; the pin wedges itself by its conical section K in housing 1 (the angle of conicity in the housing equals 20° in the pin - 15°). The wedging should eliminate backwards movement of the pin to the left and ensure the separation of the propellant valve from the solid-reactant gases after the detonation of the explosive cartridge.

Shank 3 of the valve, no longer holding the pin, shifts under the pressure of the liquid and spring. (Figure 2.21 does not show the construction of the propellant valve).

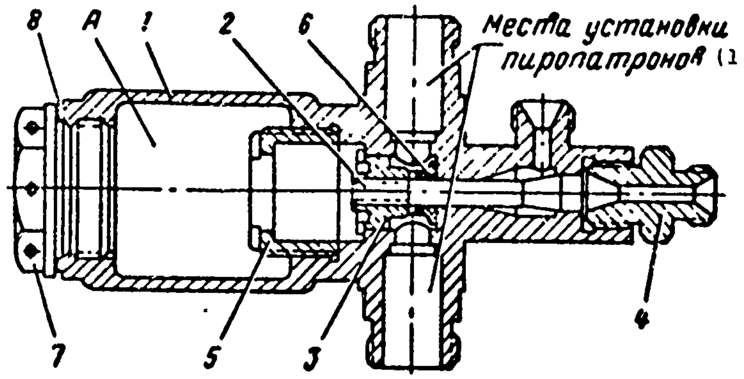

Fig. 2.20. A normally closed explosive valve with isolation of the solid-reactant gases: 1 - housing; 2 - valve; 3 - piston; 4 - connector tube; 5 - detent; 6 - stop ring; 7 - plug; 8 - gasket; A - space for solid-reactant gases. KEY: (1) Site of placement of explosive cartridges.

The seal of the liquid cavity of the valve until the triggering of the explosive cartridge is provided by the tight squeezing of bead b in housing 4 of the propellant valve using studs 9. The width of cut bead b usually amounts to 0.2-0.3 mm.

The material of pin 7 is alloys VT6 or steel Khl7N2. Housing 1 of the explosive pin and piston 5 are manufactured from aluminum alloy D16. The sealing of piston 5 in housing 1 of the explosive pin is ensured by 1 (as is shown in Fig. 2.21) or two labyrinth grooves c.

The piston is joined to the pin by means of cement ED-6. A diametric clearance between piston 5 and housing 1 is maintained with an ordinary running fit.

To avoid the penetration of solid-reacting gases into the propellant cavity of the valve during movement of the pin, seal ring 8 is provided, manufactured from teflon; the pressure of the solid-reactant gases, pressing on the seal ring, compress it against the face of the valve housing. The presence of plug 2

with gasket 6 prevents the escape of the solid-reactant gases to the atmosphere. Let us note that the absence of plug 2 will strongly accelerate the extraction of the pin; the shifting of the moving system will occur so abruptly, that sometimes the breaking off of the rod of the pin will occur.

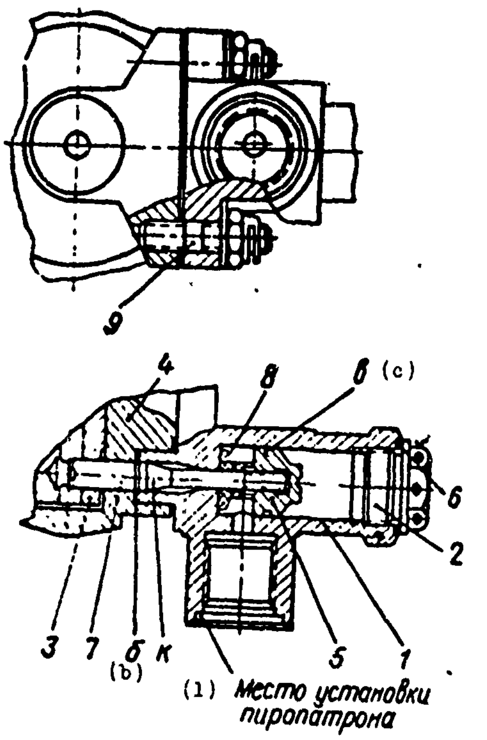

Fig. 2.21. Explosive in assembly: 1 - explosive pin housing; 2 - plug; 3 - shank of the locking mechanism of the propellant valve; 4 - propellant valve housing; 5 - piston; 6 - gasket; 7 - pin; 8 - seal ring; 9 - stud; b - bead on pin; c - labyrinth groove; K - cone of pin. KEY: (1) Location of explosive cartridge.

As a result of the experimental finishing a number of defects in the design of the pin assembly, shown in Fig. 2.21, were detected. In the majority of cases they were related to the use of fluoroplastic seal ring 8. At low temperatures the seal ring turned out to be nonhermetic and the penetration of gases into the liquid cavity occurred. This was especially noticeable during operation of the assembly with low-boiling oxidizers, when the temperature of the pin and of the seal ring was reduced to -150 to -160°C. Sometimes (during movement of the pin) the seal ring shifted together with the pin, without hinderance permitting the

solid-reactant gases to come in contact with the propellant. Such a defect is not permissible when operating with liquid oxygen. However, even at ordinary temperatures there were cases of the destruction of the seal ring by the melted residues thrown from the explosive cartridge of the metal diaphragms, which were provided by the construction of the explosive cartridge. The entry of incandescent particles to seal ring 8 led to the mechanical failure of this part.

The absence of wedging of piston 5 and its insufficiently reliable sealing from the solid-reactant gases sometimes lead to opposite movement ("recoiling") of the pin.

The presence of only two studs for fastening the pin device led to the nonuniform pressing of bead b of the pin and with the overtightening of the studs led to the destruction of the flange of the pin (at temperatures of -150 to -160°C).

The pin design shown in Fig. 2.22 turned out to be more reliable. Just as in the preceding design, the sealing of the cavity of the pin to prevent propellant from entering it, up to the moment of triggering of the explosive cartridge, was ensured by the pressing of bead b of pin 7, but this is achieved by using coupling nut 4, which has a left-hand (on the side opposite the explosive pin)and a right-hand thread. In this way the convenient assembly and reliable fastening of the explosive pin are ensured, preventing misalignment and creating a uniform squeezing of bead b of the pin.

The sealing of piston 5 from the solid-reactant gases during triggering of the explosive cartridge has been improved - this is accomplished by rubber ring 8. The cavity behind the piston is not a dead end, but has an opening, shielded by diaphragm 3, which is made from aluminum foil 0.05 mm in thickness. In case of penetration of gases through ring 8 the pressure rises, the diaphragm breaks, and the solid-reactant gases can escape outside.

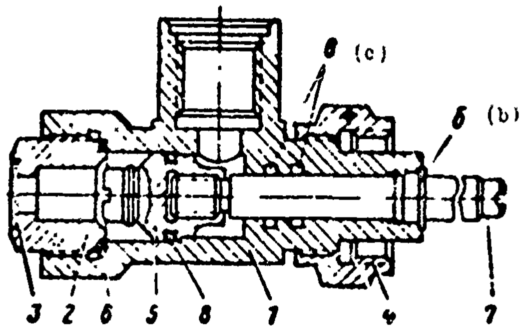

Fig. 2.22. Explosive pin: 1 - housing; 2 - detent; 3 - diaphragm; 4 - coupling nut; 5 - piston; 6 - gasket; 7 - pin; 8 - seal ring; b - pin bead; c - annular passages.

However, in practice no cases of breakthrough of gases through the piston seal were observed, although the diaphragm breaks rather frequently as a result of the compression of air during the abrupt movement of the pin.

The possibility of "recoil" of the pin is excluded in this case, since the force of the solid-reactant gases acts on the piston for a long time. Moreover, in this design, apart from the wedging of the pin in housing 1, the additional wedging of piston 5 in detent 2 is provided for. The use of a double jam permitted an increase in the angle of taper of the pin up to 30°.

The sealing out of solid-reacting gases from the propellant valve during movement of the pin is achieved by two annular channels c in housing 1. The diametric clearance between the pin and the housing is within the limits of 0.045-0.124 mm (the nominal diameter ∅ 10 mm). If a rubber ring is placed into one of the channels, then the possibility of gases entering the valve is reduced even more.

The materials of the basic parts of the pin, shown in Fig. 2.22, are similar to those employed for the pin shown in Fig. 2.21.

The manufacture of the explosive pins is rather laborious, however this is justified by the high degree of reliability of the unit's operation.

Lock-pin assemblies, similar to those described above, are put in explosive valves of various designs. A sample design of one of these explosive valves is shown below.

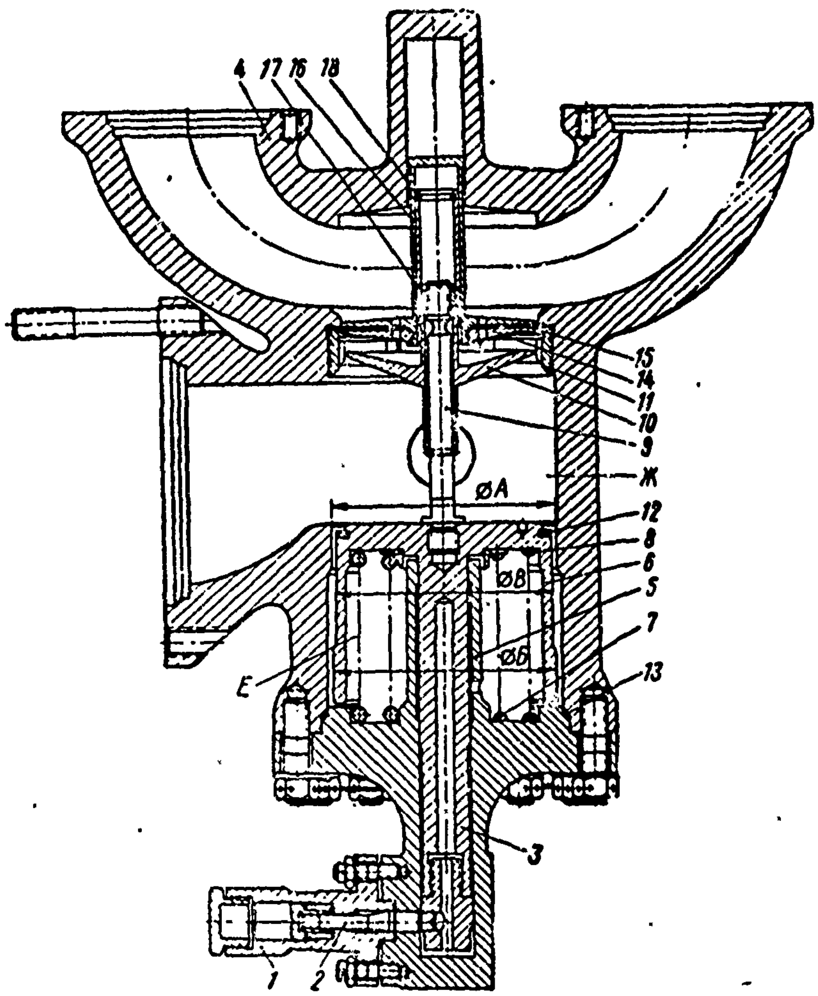

Fig. 2.23. Explosive valve of the lock-pin type: 1 - housing of pin; 2 - pin; 3 - shank; 4 - housing; 5 - cover; 6 - valve; 7, 8 - springs; 9 - explosive bolt; 10 - guide; 11 - seat; 12 - seal; 13 - rubber seal ring; 14 - disk; 15 - diaphragm; 16 - sleeve; 17 - nut; 18 - stop ring; E, Ж - cavities.

The lock-pin explosive valve, shown in Fig. 2.23, is designed for handling fuel.

With an increase in pressure at the inlet to the valve (cavity Ж) diaphragm 15, compressed between sleeve 16 and disk 14, is cut along its outer diameter, sleeve 16 is moved by the flow of propellant (in Fig. 2.23 at the top) and propellant enters the mainline behind the valve. However, as a result of the presence of the disk in guide 10, the flow-through section of the assembly is greatly blocked and the flow rate of propellant will be small, providing operation only on prestage mode.

With a growth in the pressure in front of the valve the pressure differential on the disk of guide 10 increases. At a certain (fixed) value of this differential a break occurs in the neck of explosive bolt 9. With the breaking of bolt 9 guide 10 abruptly shifts (upward), opening the flow-through cross section; the flow rate is increased, reaching a value corresponding to main stage mode.

To close the valve it is sufficient to trigger the explosive cartridge[1], installed in housing 1. With the bursting of the explosive cartridge pin 2 leaves the shank of valves 6 and the latter, moving under the action of springs 7 and 8 and the pressure of the propellant, seats itself in seat 11. The flow is cut off - the valve is closed. The required hermeticity is provided by seal 12.

To prevent too abrupt a closing of a valve, leading to strong hydroshock, the braking of the motion of valve 6 is provided for. Outer diameter Б of valve 6 is several millimeters larger with respect to diameter В (see Fig. 2.23), so that the diametric clearance Δ_1 between housing 4 along surface A and the valve is small ($\Delta_1 = \varnothing A - \varnothing Б \approx 0.5$ mm). At the start of movement of the

[1]The location of the explosive cartridge and the explosive cartridge itself are not shown in Fig. 2.23.

valve the liquid on the large diametric clearance $\Delta_2(\Delta_2 = \varnothing A - \varnothing B = 7\text{-}8$ mm) without hindrance enters the sleeve in cavity E, where the pressure is equalized to the pressure at the inlet to the valve. However, when the clearance is abruptly reduced (from Δ_2 to Δ_1) after valve 6 has traveled a certain distance, the change in pressure in cavity E will limit the speed of the valve, so that the liquid cannot enter cavity E with the former flow rate: for movement of the valve it is essential that the forces from the pressure in cavity E and from the elasticity of the springs exceed the force from the pressure beneath the valve (in cavity Ж).

With the adjustment of the assemblies for certain flow rates and pressures values of clearances Δ_1 and Δ_2 were required to be selected experimentally; since during movement of valve 6 a strong lateral force is experienced (in Fig. 2.23 - to the right) due to the unsymmetric flow of the stream, it is essential to accomplish the precise centering of valve 6 in cover 5 and the careful selection of the material of the valve for the purpose of eliminating nadirs.

2.3. EXPLOSIVE CARTRIDGES

The operating conditions of explosive cartridges in propellant valves may be very different depending on the functions performed by the assembly, in which the explosive cartridge is installed. Therefore, from the set of requirements imposed on the explosive cartridges, we will here examine only the most general, universal requirements.

The explosive cartridge should operate reliably and stably at various temperatures, under conditions of increased moisture, in a vacuum and under excess pressure, it should permit prolonged storage, preclude spontaneous triggering during transporting and vibration, and ensure the hermeticity of its internal seals.

Let us examine the simplest design of an explosive cartridge depicted in Fig. 2.24. This explosive cartridge consists of housing 1, in which explosive charge 2 is placed. The explosive charge consists of an igniter and the basic fuel mixture. Inside the igniter the hot wire is placed. The lead-out ends of the hot wire pass through ring 7 and are soldered to terminals 4. The insulation of the terminals is made using panel 6, sealed by gasket 10. The explosive cartridge is fastened using connector nut 3, which is held on the housing by stop ring 5.

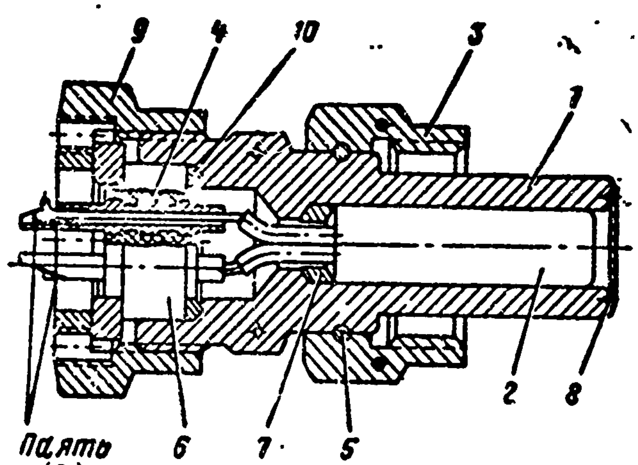

Fig. 2.24. The explosive cartridge: 1 - housing; 2 - explosive charge; 3 - connector nut; 4 - terminal; 5 - stop ring; 6 - panel; 7 - ring; 8 - membrane; 9 - nut; 10 - gasket.
KEY: (1) Solder.

The main part of the explosive cartridge, which determines its operation, is the explosive charge. The explosive charge is covered by membrane 8. The explosive composition is compressed under pressure. The ignition of the igniter (the explosion of the explosive cartridge) is accomplished by direct or alternating current.

The pressure of the solid-reactant gases, developed during the explosion of the explosive cartridge depends, naturally o

the weight of the explosive charge and on the volume of the solid-reacting gases (the volume of the gas cavity after triggering). The value of the pressure is approximately proportional to the weight of the explosive charge (all other conditions being equal). With a decrease in the volume the pressure does not increase in direct dependence upon it.

The value of the pressure, developed by the explosive cartridge with a certain explosive charge in a certain volume, depends on the temperature of the explosive charge. With a reduction in the temperature the value of the pressure is somewhat reduced, and with an increase in pressure - it increases. According to the author's data, a change in temperature from +15°C to -150°C leads to a reduction in pressure of, on the average, not more than 5-10%.

A chemical analysis of the combustion products of the majority of pyrotechnic compositions shows that these products are a reducing medium, i.e., the reaction of the combustion of the explosive composition procedes with a certain shortage of oxidizer.

The most important requirement imposed on explosive cartridges is the assurance of stability of the developed pressure (with constant weight of the explosive composition and value of the working volume). However, these pressures frequently have different values, even for one and the same batch of explosive cartridges under identical conditions.

The variation in operating pressures can be explained by the different rates of combustion of a pyrotechnic composition and by different surfaces of combustion in various samples of explosive cartridges. The difference in the surfaces of combustion is connected with the phenomenon of cracks in the explosive charge. The presence of cracks increases the burning surface.

The greater the possibility of crack phenomena, the less constant is the surface of combustion and the greater is the variation in pressures which can be anticipated. To reduce the possibility of crack phenomena the force of compression of the explosive charge is increased.

With large experimental volumes, when the gas pressure is low, there appears the possibility of the ejection of the explosive charge from the housing of the explosive cartridge. When the explosive charge burns inside the housing of the explosive cartridge, its destruction is possible, which will lead to changes in the conditions of burning and which promotes instability of the developed pressure.

The value of the pressure may also be somewhat reduced as a result of gas leakages, for example, through the panel of the lead-out terminals of the explosive cartridge or through the threaded connection of the explosive cartridge with the housing of the explosive valve.

The presence of variation in the operating pressures of explosive cartridges forces us to check the working reliability of explosive valves with explosive cartridges, which are stronger or weaker than the nominal. Successful passing of such tests guarantees reliability of the valve design.

The most important parameters of explosive cartridges are:

a) the value of the operating pressure during the test in a given volume;

b) the time to reach this pressure from the moment of supplying the current.

For explosive cartridges employed in automated assemblies of LPREs, of greatest interest is the operation in small volumes - 2-3 cm^3.

Measurements of pressure and the nature of operation of the explosive cartridges are made in special detonation chambers using crusher gauge devices and pressure sensors.

The value of the pressure created by the explosive cartridges is determined in the crusher gauge device from the value of the deformation (the shortening) of a copper column (a so-called "crusher gauge," or "a crusher-gauge column"), set up in the housing of the instrument.

The external form of the crusher gauge instrument is shown in Fig. 2.25.

For protection from the effects of the flame and high temperature after the installation of the crusher gauge the internal cavity of connector tube 2 is plugged up with a special lubricant.

The crusher gauge columns are manufactured in large batches, and with high accuracy. To increase the pressure measurement accuracy all the crusher gauge columns are first (prior to their use in the crusher gauge instruments) compressed on presses with forces corresponding to (60-80)% of the expected pressure of the solid-reactant gases.

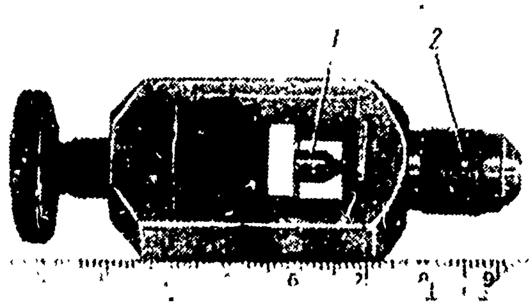

Fig. 2.25. A crusher gauge instrument: 1 - crusher gauge column; 2 - connector tube.

After this, part of the crusher gauge columns from the batch is tested for compression at stresses corresponding to the range of anticipated pressures, with a small interval between them (for example, a few pieces are tested for compression at a force of 1000 kgf, a few are tested at 1005 kgf, a few are tested at 1010 kgf, and so forth), and then the linear compression of the crusher gauge is determined. On the basis of these tests so-called "tare tables" are created (specially for each batch), in which the equivalent values of the stresses are cited for various values of deformation (shortening) of the column.

In measuring the pressure developed by the explosive cartridge in the detonation chamber, it is suggested that two crusher gauges each be set up simultaneously. The stability of readings of the crusher gauge columns is very high - pressure read-out variations for the two columns will not exceed, as a rule, (2-2.5)%. With significant differences in the crusher-gauge read outs (a difference in contraction of length of greater than 0.05 mm) the results of the experiment should be rejected.

Crusher gauge instruments can be used at test temperatures (before the detonation of the explosive cartridge[1]) of not lower than -50°C. At lower temperature the crusher gauge instruments do not operate reliably; high lubricant viscosity, small clearances inside the instrument lead to great errors. However, the crusher gauge instruments are used to determine the pressure of the solid-reactant gases even at lower temperature. For this, the detonation chamber with the explosive cartridge is separately cooled to a certain temperature, while the crusher gauge instruments, preheated to +50°C are screwed into the detonation chambers

[1]With the triggering of the explosive cartridge the temperature changes.

for (40-50) sec before triggering. Such a procedure allows us to obtain sufficiently reliable results.

It should be noted that as a result of the short-term effect of the maximum value of loading[1] (as a result of the cooling of the gases their pressure is reduced) the pressure measurement by the crusher gauge columns produces a somewhat understated value for the pressure relative to the actual pressure peak - deformation of the crusher gauges does not manage to occur.

The values of the pressure peaks, registered by tensometric sensors during the triggering of the explosive cartridges, is always (10-20)% higher than the pressure obtained using crusher gauge instruments.

With the aid of the sensors it is possible to determine by a loop oscillograph the nature of the process of triggering - the rate of the pressure rise, the ignition time delay and so forth.

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[1]The detonation chamber with the recommended GOST design has a special opening for the discharge of gases. In this case the pressure falls very rapidly.

CHAPTER 3

THE SEALS OF CONTROL CAVITIES OF PNEUMATIC VALVES

The hermeticity of control cavities of pneumatic valves with a translational-motion locking mechanism is ensured by one of the following types of seals:

a) by rubber V-shaped seal rings;

b) by round rubber rings;

c) by bellows.

Let us examine each of these types of seals.

3.1. SEAL RINGS

As can be seen from the examined examples of designs of pneumatic assemblies, the seal of the cylindrical surfaces of forward-moving parts is in many instances ensured by the rubber V-shaped seal rings (see, for example, Figs. 2.3, 2.7 and 2.10).

The shape of such seal rings has been standardized by state standard GOST 6969-54. Apart from this state standard (GOST), there are also special departmental standards. These norms encompass a smaller number of typical dimensions than the state standards; differing from the GOST in certain cases only in their manufacturing tolerances, these norms strictly establish the quality of manufacture, the conditions of application and the operation of the seal rings and the period of their storage.

Rubber seal rings, as compared with other types of contact seals (for instance, round rings), are used in low pressure (up to 200-300 at, if we judge by developed designs), and ensure better hermeticity and longer operating service life. However, they create relatively greater friction forces and have larger sizes.

In LPREs the seals operate with a translational motion speed not exceeding 3-4 m/sec.

Figure 3.1a shows the basic dimensions of the seal in the free state (prior to installation), which are controlled during the inspection of the newly manufactured seal rings. The basic dimensions include diameters D_2 and d_2 and height H. Moreover, dimensions D_1 and d_1, h and f (see Fig. 3.1b) are regulated and ensured by the manufacturing technology.

For certain types of joints the contact pressure, created at the site of contact between the soft sealing part and the shaft (housing) of the assembly during the supplying of working pressure, theoretically exceeds the working pressure of the sealed medium creating it. In these cases $p_2/p \geq 1$, where p_2 is the contact pressure, and p is the sealed pressure.

If the seal does not possess such properties ($p_2/p < 1$), then to ensure hermeticity it is essential to provide preload (clamping) of the sealing part - load until the pressure is supplied. In practice, preload over the entire diameter of the connection is always essential, if only to eliminate the possibility of etching due to manufacturing defects.

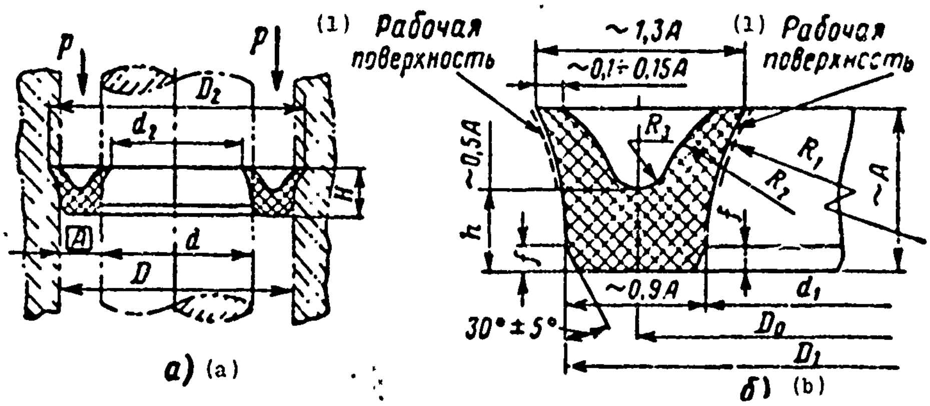

Fig. 3.1. Rubber seal ring: a - basic dimensions of the seal ring unit (A - support width, determining the working clearance between shaft and housing); b - dimensions determining the shape of the seal ring (see GOST 6969-54). KEY: (1) Working surface.

Preloading of the seal rings is accomplished using internal stresses, arising in the seal ring during its installation in the assembly. In the free state the internal diameter d_2 along the blades of the seal ring (see Fig. 3.1a) is less than the outer diameter of the shaft d, and the outer diameter of the blades of the seal ring D_2 is larger than the inner diameter of housing D. Elasticity of the rubber of the seals compensates for deviation from the proper geometrical shape of the surface of the shaft and of the housing (ellipse, conicity and so forth), which can arise during manufacture. When pressure is supplied in the direction of arrow p (see Fig. 3.1a) the clamping force of the seal ring against the shaft and housing increases even more.

The seal ring may be assembled on the movable rod (see, for example, Fig. 2.3 - seals 16 and 17) or in the stationary housing (see Fig. 2.3 - seal 15).

When the rod makes contact with the inner blade of the seal ring the theoretical ratio of pressures p_2/p is within the limits of 1.0026-1.01 (see work [9]), i.e., even without preload in this connection an excess of contact pressure above the working pressure is ensured.

In the contact of the housing with the outer blade of the seal the ratio p_2/p is theoretically 0.9851-0.9975, i.e., in this case preload is essential.

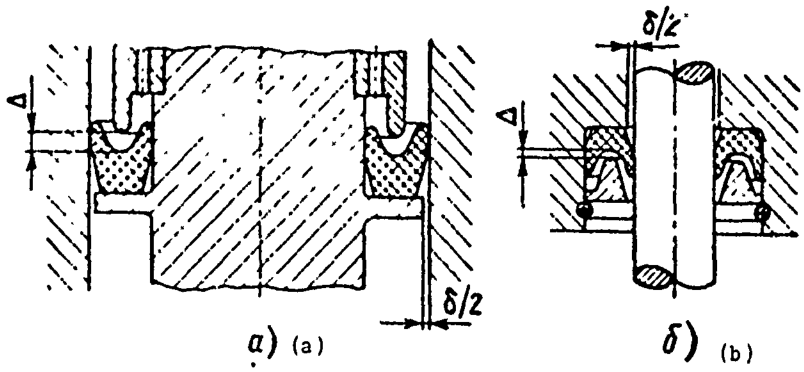

Fig. 3.2. Diagram of seal ring installation: a - of a movable seal ring; b - of a stationary seal ring.

A diagram for the installation of seal rings is shown in Fig. 3.2.

Proceeding from the above cited considerations it is desirable that the seal ring be fixed (fastened in the housing), so that the movement of the part takes place on the inner blade, since in the stationary connection hermeticity is achieved more easily.

In selecting the brand of rubber for the seal rings one should take into consideration the frost resistance of the rubber (i.e., the temperature at which elasticity is still maintained), as well as the resistance of the rubber to a given propellant (both to a liquid as well as to its vapors).

Rubber seal rings cannot be used for valves which handle low-boiling liquids (liquid oxygen, nitrogen, etc.), since in this case the temperature is significantly lower than that at which the rubber becomes brittle. Rubber seal rings are also not used in propellant cavities of assemblies which handle nitric acid or other oxidizers similar to it in their properties.

Practically speaking, for oxidizer valves rubber seal rings are employed only in assemblies which handle hydrogen peroxide.

Rubber seal rings are much more widely used in fuel valves, although fuels based on petroleum products (kerosene, turpentine and others) to one degree or another cause the swelling of many brands of rubber.

Seal rings which handle compressed gases - air, nitrogen, helium - and also alcohol and hydrogen peroxide employ rubber on a base of methylstyrene (SKMS-10) and sodium-butadiene (SKB) rubbers. Such seal rings are capable of existing under the effects of the cited liquid propellants and their vapors continuously for not less than 10-12 days without change in their properties. For seal rings which handle kerosene rubber based on butadiene-nitrile rubbers (SKN-26 or SKN-26+SKN-18) is recommended.

With the swelling of the rubber it becomes significantly softer than it was prior to coming into contact with the fuel, it looses its elasticity, and the seal rings start to expand in the inner and outer diameters[1]. A swelled seal ring taken from the propellant after being in air several days gradually assumes its former dimensions, since the propellant which penetrated into the rubber evaporates.

When the seal ring is in the assembled unit, then the swelling of the rubber causes a significant increase in the interference in the inner and outer diameters of the seal, which increases the stresses in the rubber (of course, the dimensions of the seal rings in the assembled unit do not change, since the rigid

[1]With a nominal outer diameter of 60-70 mm an increase in diameters may amount to 1-2 mm.

walls of the housing or of other parts prevent this). With the swelling of the rubber pores and minute cracks, which were earlier insignificant and invisible to the naked eye, become deeper, expand and give rise to the failure of the rubber.

The rubber of the supporting part of the steel ring (it becomes softer from contact with the fuel) easily flows even into the small clearances δ between the support and the housing (or the support and the shaft - in the case of stationary seal rings, see Figure 3.2b) and in these clearances it cracks and is torn. Its surface, which was shiny prior to contact with the propellant, becomes dull, rough, and torn.

The hermeticity of seal rings swollen under the action of propellant do not deteriorate immediately; on the contrary, sometimes as a result of the increase in interference the hermeticity is even improved.

However, after prolong contact (on the order of several days) of the rubber of the seals with the propellant, under the action of even very slight pressure the propellant begins to diffuse, to leak through the thickness of the rubber. Propellant which has penetrated through the "liquid" seal rings (sealing the liquid cavity of the assembly) enters the support section of the "air" seals, which seal the control cavity, and has a very adverse effect on their efficiency. The entry of drops of propellant into the air seals gives rise to their unequal swelling and as a result of this to deformation; the seal ring loses its given shape (geometric form) and as a result of its deformation the interference along the blades of the seal ring is reduced.

Such a phenomenon leads to loss in hermeticity of the "air" seal ring. The nonhermeticity of "air" seal rings occurs especially frequently after the holding of the assembly under

the propellant without pressure in the controlling cavity[1]. A similar combination of conditions (the presence of propellant in front of the seal rings of the liquid cavity and the absence of pressure in the control cavity) is completely possible in normally closed valves of the discharge type. Thus, until the moment of start of the engines, i.e., until the moment of the beginning of actual operation of the valve, its hermeticity can deteriorate. Therefore, under similar conditions it is essential to take measures to protect the seal rings of the control cavity either by means of a well thought-out vent behind the "liquid" seal ring, or by the installation of special reflectors or additional rubber seals which will contain the vapors or drops of propellant.

For example, in the assembly depicted in Fig. 2.3 there is installed additional protection seal ring 16, the unique purpose of which is to protect seal ring 17 from the effects of the propellant (even for a short time). The deformation of seal 16 is not of importance, however the penetration of propellant through it to seal 17 is, practically speaking, eliminated, since the quantity of propellant which seeps through seal 15 is extremely small.

In view of the effect of the propellant on the rubber, seal rings which come in contact with propellant cannot be used again if the valve is disassembled, and are replaced with new ones. For test bench assemblies a limited service period is established before overhaul. In view of all this, in test bench assemblies which handle liquid propellants (except, perhaps, alcohol) one should avoid the use of rubber seal rings in the liquid cavity.

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[1]When there is pressure the seal ring is more resistant to changes in shape.

The swelling of the seals under the effect of the propellant leads also to an increase in the friction force, to frictional instability and to a worsening of the movement smoothness of the moving system.

To avoid the mentioned phenomenon of swelling, the working capacity of seals installed in both propellant and in control ("air") cavities is determined by one and the same factors. It is natural that the hermeticity of the "liquid" seal rings, with all other conditions being equal, will be higher than the hermeticity of the "air" seal rings, since the viscosity of the propellant is considerably higher than the viscosity of the gas. However, it should be noted that as a result of the high wetting capacity of kerosene the hermeticity of the seal rings which handle this propellant will be achieved with more difficulty than with those handling other propellants with a similar or lower viscosity.

A whole list of factors influence the sealing properties of a seal ring and its efficiency.

3.1.1. Shape of the Seal Rings

The shape of the seal rings (for given diameters of the rod and housing) determines the value of the clearance of the blades of the seals, and influences the force of adhesion of the seal ring to the sealing surfaces and the magnitude of the contact pressure. An increase in the interference favors the assurance of hermeticity, especially at low pressure, but excess interference entails an increase in friction. The seal ring must not adhere to the surfaces of the shaft and housing with its support section, since in this case the friction increases sharply and the possibility of reversing the seal ring is created. It is therefore important that the support part of the seal ring have adequate rigidity and stability, but its blade should possess sufficient elasticity.

On the strength of these considerations the shape of seal rings was created according to GOST 6969-54 and allowances for deviation in dimensions were made. Seal rings of another shape, apart from that mentioned in GOST 6969-54, are not employed in propellant valves of LPRE (in contrast to EPV and gas pressure reducers, where seal rings of the "simplified" shape are frequently encountered).

The basic dimensions which characterize the stability of a seal ring with a given rod diameter are the height of the seal H (see Fig. 3.1a) and the support width A, determined by the half-difference of the inner diameter of housing D and of outer diameter of the shaft d.

The stability of the seal ring is reduced with a decrease in dimensions H and A with a fixed dimension d.

Tolerances for the diametric dimensions of the seal, housing and shaft determine the divergence in the values of the interference of the blades of the seal ring. In view of the elasticity of the blades it is difficult to examine separately interferences on the inner and outer diameters and it is possible to only estimate the total interference. Therefore, considerations apropos of the fact that interference is essential on the outer blade, while on the inner one it is not so obligatory, are rather of a theoretical than of a practical nature.

3.1.2. The Ambient Temperature

With the reduction in temperature (especially below -20°C) the hermeticity of the seal rings (chiefly at low pressures) worsens, because the force of the pressure due to the elasticity of the rubber decreases sharply (even with relatively frost-resistant rubbers). The rubber becomes harder and more rigid, and even the graduation mark on the surface of the housing or

shaft or a small foreign particle on the edge of the blade of the seal ring can effect the hermeticity to a significantly greater degree than at normal temperature. Practice shows that a reduction in temperature affects the hermeticity on the outer blade of the seal ring to a greater degree than on the inner.

3.1.3. The Quality of the Surface of the Seal Rings and of the Mated Parts

Defects on the surface of the seal rings - gas bubbles, roughness, extraneous inclusions - worsen the hermeticity and sharply decrease the operating service life. Especially important is the condition of the edges of the seal blades. They must be even, without tears, breaks, cuts and so forth. Strings on the seal ring blades frequently occur as a result of careless assembly or disassembly of the unit - when putting on and removing the seal ring. Therefore, during assembly and disassembly of the unit mounting fixtures should be used (see, for example, Fig. 3.3). Breakdown of the rubber which has begun (due to the presence of a defect) progresses rapidly when the assembly is operated.

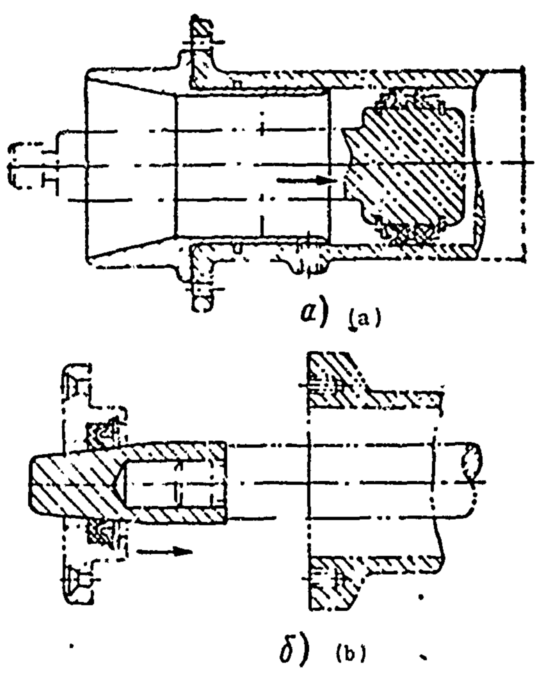

Fig. 3.3. Devices for installing seal rings: a - into the housing; b - on the rod (the parts of the assembly are shown by the dot-and-dash line.

During assembly one should not permit expansion of the inner diameter of the seal ring (even for a short time) of more than 40%.

The hermeticity is affected by the quality of the finish of the mated parts, which move relative to the seal ring. The cleaner their surfaces, the higher the hermeticity. The surfaces against which the seal ring moves are finished with a purity of ∇7 and are polished; the surfaces which only come in contact with the seal rings are finished with a purity of ∇6. An increase in the surface finishing purity not only favors an increase in hermeticity of the seal ring, but also leads to a decrease in the friction force and to an increase in the smoothness of movement.

To increase the purity of the surface of aluminum parts, these surfaces are subjected to "hard" anodizing[1]. A "hard" or "deep" anodizing produces a hard, rather thick anodic film (up to 0.1 mm), which is subjected to further mechanical treatment - grinding or polishing. After treatment the film thickness is 0.06-0.07 mm.

3.1.4. Type and Quantity of Lubrication

At low temperature the type of lubricant has a great effect on the hermeticity and the friction magnitude of the seal ring. A nonfrost-resistant lubricant solidifies and roughens the surface over which the seal ring will move; with an excess of lubricant the protuberances and nonuniformities become more significant.

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[1]Ordinary anodizing of aluminum parts produces a porous, rough surface.

Table 3.1.

Brand of lubricant	Excess pressure, corresponding to the start of movement of the rod, which characterizes the shifting effort (friction force)		Remarks
	$t = -40°C$	$t = 20°C$	
TsIATIM-221 (GOST 9433-60)	22 at	20-23 at	During assembly witnout seal rings the pressure of the beginning of movement at $t = -40°C$ equals 18-20 at (overcoming the spring pressure)
TsIATIM-205 (GOST 8551-57)	40 at	20-23 at	
Without lubrication	48 at	20-23 at	

Remarks. A substantial pressure change for a given assembly can be explained by a low value of the effective area on which the pressure acted.

In testing assemblies with seal rings lubricated by various brands of lubricants the values shown in Table 3.1 were obtained for the pressure of the start of movement of the moving system of the valve, characterizing the shifting efforts of the seal rings.

At normal and elevated temperatures the type and quantity of lubricant has a lesser effect on the operation of the seal rings.

The quantity (volume) of lubricant which can be applied to seal rings and mated parts is specified by the technology.

3.1.5. The Radial Clearance, the Operating Service Life

The magnitude of clearance δ (see Fig. 3.2) has an effect on the working capacity of the seal rings.

During the effect of high pressure in clearance δ between the support and housing (for moving seal rings) the support section of the seal ring can be extruded. When the pressure is released the rubber usually returns. After many triggerings (repeated application and discharge of pressure) as a result of this extrusion the rubber is gradually destroyed. The shiny and smooth surface becomes rough and torn. The greater the clearance δ, the more intensive is the destruction of the rubber. The clearance δ should not be made greater than that value, which is produced by fit X_3.

If necessary to ensure long service life of the seal ring (for example, in test bench valves) and at pressures exceeding 150 at we recommend that beneath the support section of the seal rings support rings made from teflon be placed, with very small diametric clearances or even with interference (for diameters of up to 30 mm and interference of up to 0.095 mm, and with a diameter of 50-80 mm - an interference of up to 0.135 mm . The thickness of the rings is approximately 2.5 mm.

The installation of the teflon rings does not cause any increase in friction; on the contrary, at high pressures (300 at) the presence of the support rings made from teflon results in a decrease in friction as a result of the elimination of the extrusion of the end face of the seal rings into the clearances.

The required value of the triggering service life of the assemblies of LPREs is ordinarily ensured by seal rings without

special care; the influence of the number of triggerings manifests itself only when there is inadequate lubrication, large clearance δ, poor quality surface of the seal or low purity of finish of the parts which move relative to the seal.

The axial play of the seal Δ (see Fig. 3.2) is selected within the limits of 0.5-1.3 mm, and with a height H greater than 6 mm - within the limits of 0.5-2 mm. The magnitude of this play in a well designed sealing unit has no effect on the operation of the seal ring. The complete absence of play Δ or an excessively large value for Δ should be avoided.

3.1.6. The Shelf Life

Of great significance for the working capacity of a seal ring is the storage period of the rubber parts.

In the storage process a rubber seal ring, just as every rubber part, "ages". The "aging" of rubber substantially reduces its elastic properties and leads to a decrease in the force of adhesion of the seal ring blade to the housing and to the rod of the assembly. Seal rings which have been kept for a long period and which have aged have considerably worse hermeticity indexes than new seal rings; this has an especially strong effect at low temperatures when the elastic properties of rubber are lower even without this. As a result of this the problem of the aging of rubber gets great attention - sometimes it is the decisive factor in solving a problem of selecting a design of one or another unit.

But how do we explain the deterioration of the elasticity of rubber with the passage of time?

Basic rubber is a high-polymer product - natural rubber, to which are added in relatively small quantities various substances (ingredients) - fillers, antioxidizers, vulcanizers, etc., which impart certain properties to the rubber after vulcanization.

The high-elastic properties of natural rubber (of any composition) are determined by its molecular structure - large long molecules and high mobility of its individual segments. The structure of a molecule of natural rubber determine the physico-mechanical indexes of the rubber. A change in the structure of the molecule (chemical transformation of the substance) the rupture or modification of the polymer chain, the formation of another bond between parts of the molecule - lead to a basic change in the physical and physico-mechanical parameters of the rubber. Such chemical changes also occur in rubber with the passage of time (the process of aging).

Thus, the aging of rubber is a consequence of various chemical reactions between the molecules of the natural rubber, the ingredients of the rubber and oxygen of the environment, accelerated under the action of increased temperature, sunlight, mechanical stresses, and leading to irreversible changes in the mechanical and physical properties.

Ordinarily during the initial storage period oxidation of the rubber occurs at a retarded rate due to the presence of antioxidants, then after a considerable time interval (which basically decides the entire permissible storage time for the rubber) oxidation proceeds with a certain constant rate, and toward the end of the process oxidation may be accelerated as a result of the catalytic activity of the reaction products during consumption of the antioxidants.

The aging of rubber under the action of various factors occurs according to various laws - on the basis of different chemical processes. Therefore, the antioxidants which are introduced in rubber, effective for one of these processes (occuring for example, under the action of light), may be barely effective during aging caused by other factors, for example, by repeated cycles of deformation.

The degree of aging is affected by the value of the stresses in the rubber: the greater the rubber is loaded, the more energetically will the aging processes take place. Therefore, seal rings which are seated with large interferences, or rings which have a large percentage of compression (see below), age more quickly than those same seal rings or other rings, assembled with less interference.

In order to predetermine the working capacity of a part during the entire storage period methods of "artificial" (accelerated) aging with subsequent checking of the working capacity of the unit are used.

The method of accelerated aging consists in holding assemblies with the investigated rubber components (seal rings, other rings, etc.), and also separately taken rubber components, at elevated temperature (50, 70 and 90°C). With the increase in temperature the processes which occur in rubber during aging take place significantly faster.

The conversion factors which determine what kind of storage period at normal temperature, equal to 20°C, corresponds to storage for several days at a temperature of 50, 70 or 90°C, depend on the brand of rubber and on the value of stresses in it. For orientation we can call the value of coefficients 11-25 for

50-150, and for aging at 90°C - the coefficients 180-750. Thus, a day of storage at 90°C corresponds to storage at a temperature of 20°C for a period of from a half year to two years.

For the characteristics of various brands of rubber in relation to their tendency toward aging the so-called "aging factor" is introduced; this is the ratio of the value of relative elongation after aging for 240 hours at a temperature of 70°C to the value of relative elongation prior to aging.

The aging factor of brands of rubber in use ranges from 0.4 to 0.7. A high aging factor indicates good maintenance of the elastic properties of the rubber; it indicates that it will be subjected to aging relatively little. However, there is no strict correspondence between artificial and natural types of aging. Therefore the aging factor may characterize only the relative properties of rubbers of various brands under some preselected conditions.

Many times attempts have been made to replace rubber in seal rings by other more resistant materials. Of great interest from this viewpoint is teflon. Teflon is a very resistant substance in a chemical respect; it is heat resistant and frost resistant. However, teflon cannot be used for the manufacture of ordinary seal rings of the V-type due to the high linear expansion coefficient and the low elasticity of this material (it flows easily).

Ordinary seal rings manufactured from teflon cannot tolerate significant temperature fluctuations, and cannot operate under low pressures. When operating in a medium with elevated temperature (+50°C), the seal ring expands considerably more than the metal of the construction, and so at great pressure the material

of the seal ring flows into the clearance (see Fig. 3.2) and foliation and cracking of the material takes place. At low temperature (-50°C) the seal ring is compressed, and sometimes even broken. During cyclic temperature fluctuations (-50°C) - (+50°C) - (-50°C) a V-shaped seal ring made from teflon usually malfunctions very rapidly.

The hermeticity of similar teflon seal rings even at normal temperature and under excess sealing pressures below 5-6 at, as a rule, is unsatisfactory in view of the weak pressure of the blades of the seal ring against the walls of the housing and shaft.

Figure 3.4 shows a subassembly of the unit, which works in the environment of an aggressive propellant, precluding the use of rubber seal rings. An attempt was made to use fluoroplastic seals with a release device of the Belleville washer type, as is shown in the diagram. However, this design is not warranted - we did not manage to provide the required hermeticity.

In order to use teflon for the seal rings a special fitting is sometimes made from spring sheet steel, which is surrounded by a circle of teflon - thus reinforced seal rings are obtained. However, such a design is not widespread.

Abroad, seal rings of the "collar" type are acclaimed, manufactured from a material similar to PTFCE, and having a thickness of about 0.5 mm.

3.2. RINGS

Round rubber rings are used for sealing pneumatic assemblies. Similar rings are employed in assemblies of on-board aircraft systems. Rubber rings, which work in the hydraulic systems of aircraft, are employed in moving connections, which operate

at pressures of 350-500 at, and in certain cases also at pressures of up to 1000 at (see work [2]), with a translational movement speed of up to 6 m/sec.

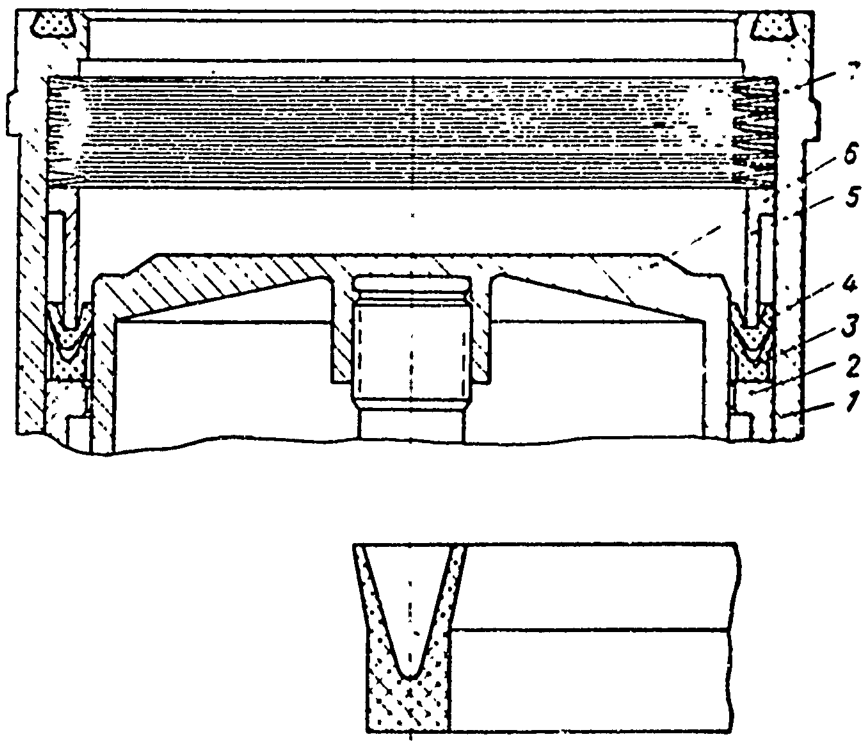

Fig. 3.4. Subassembly with mechanical release device of a fluoroplastic seal ring: 1 - valve; 2 - seal ring support; 3 - fluoroplastic seal ring; 4 - release clamp (plastic or metal); 5 - bushing; 6 - sleeve; 7 - Belleville washer.

The advantage of seals which use rubber rings are their small dimensions, relatively low frictional value and maximum design simplicity. They are employed in the pneumo-automatic assemblies of LPREs within the temperature range of $\pm 50°C$.

Sealing by rubber rings has until now been used mainly for the hermetization of the controlling (air) cavity. For the sealing of a liquid cavity rubber rings are usually not employed

because of the undesirable action of the propellant on the rubber and the possibility of swelling of the rubber.

For the manufacture of seal rings rubber which possesses high frost-resistance and high elastic properties is employed. The most widely used rubbers are those based on methylstyrene rubber SKMS-10, employed for sealing compressed air.

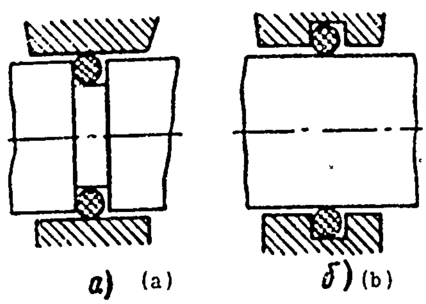

Fig. 3.5. The sealing of forward moving parts by round rings; a - seal with moving ring; b - seal with stationary ring.

Rubber rings may be installed both on the forward-moving parts themselves and on the stationary parts mated with them; sealing is affected both on the inner and outer diameters of the ring (Fig. 3.5).

Certain investigators (G. V. Makarov [9] et al.) think that higher contact pressures (a higher ratio p_2/p = 1.0-1.07) and, consequently, more reliable hermeticity are obtained by sealing with stationary rings on a moving rod. Rings fastened on a rod and moving together with it provide a somewhat lower value of p_2/p, equal to (according to G. V. Makarov) 1.0-0.82.

In several earlier works it was assumed that the contact pressures along the inner and outer diameters of the ring were equal.

To ensure the hermeticity of the connections the rings must be installed with a preload which provides preliminary contact pressure on the sealing surfaces[1]. The value of the preload is determined by the compression of the ring.

The working capacity and the hermeticity of the rings are determined by the factors enumerated in sections 3.2.1-3.2.5.

3.2.1. Degree of Compression of Rubber

The compression of the ring w (in percent) is determined as a ratio of the difference in height of a section of the ring before and after the installation of the connection to the diameter of the section of the ring prior to compression (Fig. 3.6a):

$$w=\left(\frac{d-l}{d}\right)\cdot 100\%, \text{ or } w=\left(1-\frac{l}{d}\right)\cdot 100\%.$$

Too great a value of w (i.e., excessive compression of the ring) will result in an increase in friction at low pressures, in the breakdown of the rubber and will cause accelerated aging of the rubber. Too little compression is not permissible because as a result of the deformation of the unit or due to the presence of eccentricity of the mated parts on a certain part of the circumference of the ring compression of the rubber will be completely absent, resulting in the loss of hermeticity. Nonuniform and too little compression of the ring often leads to the turning (cranking) of the ring in the groove, accompanied by the reversal of the ring or the deformation of its transverse cross section, which also leads to nonhermeticity.

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[1]The contact pressure created by preloading of the rings significantly increases the contact pressure existing at the seal rings.

Evaluating the magnitude of compression of the rubber, one should keep in mind that in the installation of the ring into the groove deformation of the ring cross section occurs, since the ring is being seated onto the shaft (see Fig. 3.6b) with a certain interference ($d_B - D_H$). It is recommended that the interference value

$$\varepsilon = \frac{d_B - D_K}{D_K}$$

be maintained within the limits of 1-5%.

As a result of this interference the cross section of the ring is reduced somewhat, and it ceases to be round and becomes oval, as a result of which the compression value w calculated using the above given formula is in practice reduced. With large relative interference ε the flattening on the ring should be taken into consideration when determining the degree of compression w. Therefore the following expression will be more precise:

$$w' = \frac{b - l}{b},$$

where b is the height of the ring, seated on the rod. (The expression $w = \frac{d-l}{d}$ is more suitable for estimated calculations).

Flattening of the ring increases with the rise in hardness of the rubber (the oval becomes more pronounced).

Prolonged storage of the unit with low compression of the ring can lead as a result of aging (with the loss of the elastic properties of the rubber) to a decrease in or to the total elimination of the interference.

All the mentioned factors are taken into consideration when selecting the value of compression of a rubber ring. Ordinarily for moving connections the compression of rings with a channel of type I (Fig. 3.7) amounts to 15-25%[1] (if necessary to reduce

[1]For stationary connections the recommended compression with a groove of type I amounts to 25-50%.

friction, sometimes it is permissible to reduce the compression up to 10%.

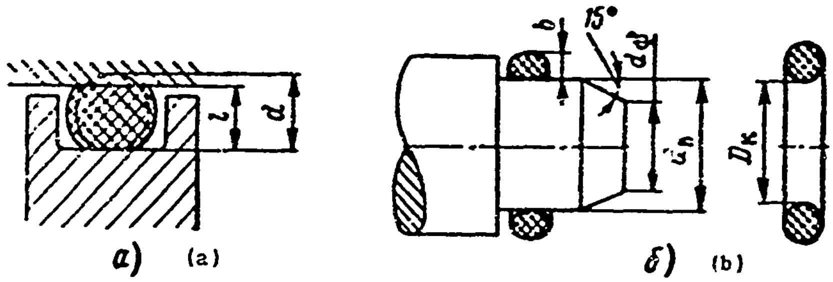

Fig. 3.6.

The ring with cross sectional diameter d is manufactured with an accuracy which does not exceed ± 0.15 mm. This variation in the dimensions of the cross sectional diameter of the ring "eats up" the greater portion of the tolerance for compression, as a result of which the dimensions of the channel beneath the ring and the diameter of the mated part must be maintained with high accuracy. For example, with a diameter of the ring cross section equal to 3 mm, only as a result of the tolerance for the diameter of the cross section of the rubber ring equal to ± 0.15 mm, the compression of the ring (nominal 15%) may vary within the limits of 10.5-19%.

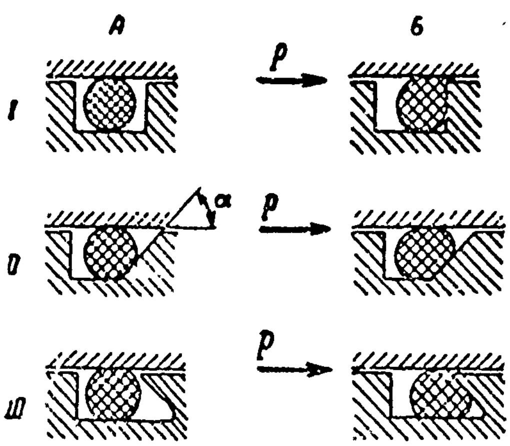

Fig. 3.7. Types of grooves for steel rings: A - ring prior to pressure delivery; Б - ring after pressure delivery.

The larger the diameter of the connection, the more difficult it is to maintain the required degree of compression of the rubber, since the value of the required tolerance for compression (in mm) does not depend on the diameter of the connection, while the manufacturing accuracy of the metal parts is determined precisely by the magnitude of the diameter. An increase in the diameter of the rubber ring cross section permits a decrease in the variation in the percentage of compression.

Recommended values for the ring cross section d depending on its inner diameter $D_н$ (see Fig. 3.6) are shown in Table 3.2.

Table 3.2.

d, мм	2,5	3,0	3,5	4,0	4,5	5,0
$D_к$, мм	до 18	18÷34	36÷48	50÷120	125÷190	Свыше 190

The diameters of the seats under the ring are usually made with plain-fit tolerances.

To reduce the deformation of the ring while installing it on the rod (deformation of the inner diameter must not be greater than 30%) chamfers should be made on the rod with an angle of 15-30° (see Fig. 3.6b). The difference ($D_к$-$d_ф$) amounts to approximately 1 mm. There should also be similar chamfers for ease of assembly on the housing, where the rod enters with the ring.

3.2.2. The Type of Groove

There are several types of grooves for steel rings (see Fig. 3.7).

In all cases, the cross sectional area of the groove for a rubber ring must be clearly greater than the cross section of the wing itself.

With a groove of type I the ring works identically with a supply of pressure both from the left and from the right. The axial clearance of the ring with a groove of this type should be from 0.5 mm (using a ring with a cross-sectional diameter of 2 mm) up to 1.5 mm (using a ring with a cross-sectional diameter of 5 mm). It is recommended that the cross-sectional area of the groove amount to ~1.3 of the cross sectional area of the ring.

With a groove in the housing made according to type II the ring operates during the delivery of pressure from the left side. Under the effect of the pressure the rubber is forced into the conical slot and ensures a reliable seal. With a nominal cross sectional diameter of the ring equal to 3 mm, the width of the groove l is made equal to ~6 mm.

The hermeticity of the seal with a type-II groove is the greater, the higher the sealing pressure. Similar seals are employed in the moving joints very rarely - only with unilateral pressure delivery and only in those cases where during the delivery and removal of pressure it is not essential to ensure smoothness of movement of the rod relative to the housing, since with this unit's design the friction forces are great and variable. Such a seal is widely employed in fixed connections.

The permissible compression of the rubber (in mm) with a type-II channel is somewhat less than with a type-I channel.

A type-III channel is used for fixed connections when supplying pressure only from the left side. Such channels ensure minimal friction force, since with an increase in pressure above a certain limit as a result of the deformation of the ring (it is pressed

into the undercut of the groove) the contact area is reduced for the rubber ring, which sits in the channel of the moving rod, with the surface of the cylinder. For this reason the hermeticity of the joint made with a type-III channel will be disrupted with an increase in pressure at a lower pressure, than the hermeticity of a joint with grooves of type I or II. To provide reliable hermeticity the length of the lines of contact of the rubber ring with the surface of a cylinder must be no less than 25% of the cross-sectional diameter of the ring.

The greater the value of angle α (see Fig. 3.7), the lower the friction but also the lower will be the value of the sealing pressure permissible with respect to the hermeticity. With a pressure of up to 150 at the maximum value of angle α equals 45°.

3 2.3. Diametric Clearance

Of great significance for the working capacity of rings in type-I channels is the value of clearance Δ (Fig. 3.8a), into which the rubber is pressed. The pressing of the ring into the clearance is the main reason for the wear of the rubber and determines the ring's service life.

The breakdown of the ring begins, as a rule, at point A (see Fig. 3.8a), at the site of maximum stresses in the rubber. The breakdown of the ring leads to a loss in hermeticity. The cross section of the destroyed ring is depicted in Fig. 3.8b. The pressing of the ring into the clearance occurs the more intensively, the higher the value of the pressure, the greater the clearance and the lower the hardness of the rubber.

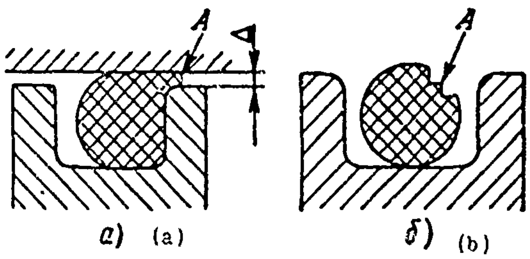

Fig. 3.8. The wear of a seal ring: a - pressing of a ring into clearance Δ (at point A - the beginning of destruction of the ring); b - cross section of the destroyed ring.

Professor T. M. Bashta recommends the use of the clearance values shown in Table 3.3 for rings (manufactured from rubber with a hardness equal to 70 Shore units), depending on the pressure (operating in liquid).

Table 3.3.

Pressure, at	0-40	40-100	100-200
Side clearance, mm	0.2-0.1	0.1-0.06	0.06-0.02

Since for assemblies of LPREs the service life is significantly less, and the conditions of their operation are easier than in aircraft power plants, then for the valves in LPREs the magnitudes of clearance can be increased. The values of the clearances are designated, based on the possible deformation of the unit, the overall accuracy of manufacture of the assembly and the diameter of the connection. Ordinarily, a fit X_3 is employed (with a speed of forward movement of about 0.5 m/sec).

3.2.4. The Sharpness of the Channel Edges

Rounding of the edges of the channel (an increase in the radius of rounding at point A, see Fig. 3.8) promotes the entry under the action of pressure, of the rubber into the clearance and, consequently, increases the wear of the ring. Therefore the

radius of rounding must be minimal - on the order of 0.02-0.05 mm. Sharp edges and projecting edges are also dangerous, since they can lead to cuts in the rubber.

3.2.5. Quality of Manufacture of the Ring

In order to increase the reliability, rings made from rubber cord, butt-cemented, are almost not used; rather, rings manufactured in a special die-casting mold are employed. The die-casting mold, as usual, consists of two halves (Fig. 3.9). In the stamping of the ring at the site of contact of the two halves of the mold a small fin of rubber is formed. It is important that this fin not be on the surface of the ring at the sealing site, as is shown in Fig. 3.9b. It is therefore recommended that the mold be made at an angle of 45-60° to the plane of sealing (see Fig. 3.9a), but even under these conditions a large fin size can sharply worsen the hermeticity of the unit, especially when the ring turns. It is therefore essential to control the value of the fin. A fin value exceeding 0.3 mm is not permitted.

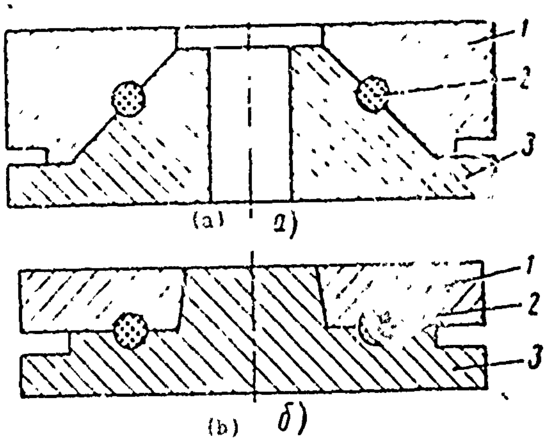

Fig. 3.9. Diagrams of the joint of the molds for rings: a - correct location of the joint; b - not recommended location of the joint; 1 - upper plate; 2 - pressed ring; 3 - base of the mold.

Other factors - such as the temperature of operation, degree of finish of the parts, storage period - affect the rings just as they do the seals. The quantity of lubricant affects the operation of the rings less than the operation of the seals.

Figure 3.10 shows the dependence of the force of friction on the pressure with sealing by rings and seals. As can be seen from Fig. 3.10, the rings produce substantially less friction than the seals. However, the friction forces are still in this case substantial and, most importantly, unstable.

The friction of the rings depends on the rubber pressing force against the surface of the part. The value of this force at low pressure is determined by the degree of compression of the rubber. With an increase in pressure of the working medium the value of the contact pressure increases in proportion to this pressure. At high pressures the value of compression does not influence the value of the shearing forces.

Figure 3.11 shows the dependence of the coefficient of friction of the ring in a type-I channel (see Fig. 3.7) on the pressure (according to the data of T. M. Bashta) while handling a liquid. Although the coefficient of friction decreases with an increase in pressure, nevertheless the friction force itself rises with an increase in pressure.

Figure 3.12 shows the dependences of the pressure of the initial shearing forces of moving (see Fig. 3.12a) and nonmoving (see Fig. 3.12b) rings (after a 5-minute holding under pressure). Rings of one and the same typical dimension, with a cross-sectional diameter of 3 mm were subjected to testing. For moving rings (fastened on a piston) the diameter of slip was equal to ~76 mm, for stationary rings (fastened to the cylinder) the diameter of slip amounted to ~70 mm.

The shearing force of the rings significantly exceeds (by 2-3 times) the friction force during movement. The friction of the rubber while dragging from the site after prolonged rest is higher than during repeated shifting, which occurs immediately after the first movement. In the stationary state displacement of the lubricant from the point of contact of the ring and of the mated part takes place. "Adhesion" of the rubber also occurs.

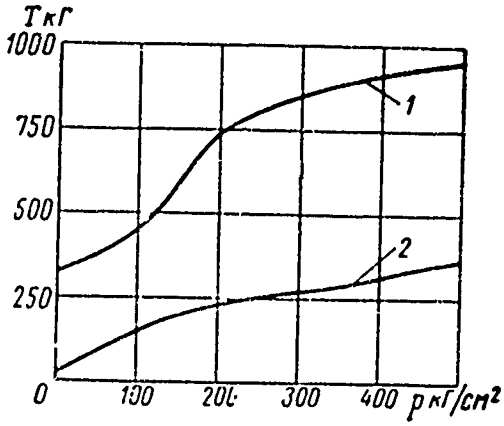

Fig. 3.10. Dependence of the friction force on the pressure in sealing by rings and seals (for one and the same shaft diameter): 1 - rubber seal (GOST 6969-54); 2 - two seal rings of round cross section.

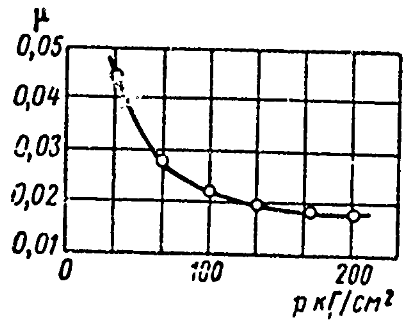

Fig. 3.11. Dependence of the coefficient of friction μ of a rubber ring on the pressure (grooves of type I, see Fig. 3.7).

The friction force of a ring increases with a decrease in temperature from -15°C to -40°C. This is apparently connected not only with the change in hardness and properties of the rubber, but also with the deterioration of the antifrictional properties of the lubricant.

Friction and wear of rings are reduced with an increase in the surface finish of the part, relative to which the ring moves; the wear of rubber operating against steel is less than during its operation against aluminum[1] or against stainless steel. Chrome-plating of the surface of the part reduces the wear of the ring. The surface mated with the ring is ordinarily finished with a purity of ∇7, and the groove - with a purity of ∇6. To reduce friction the groove is filled with heavy oil (if this is possible with respect to the operating conditions).

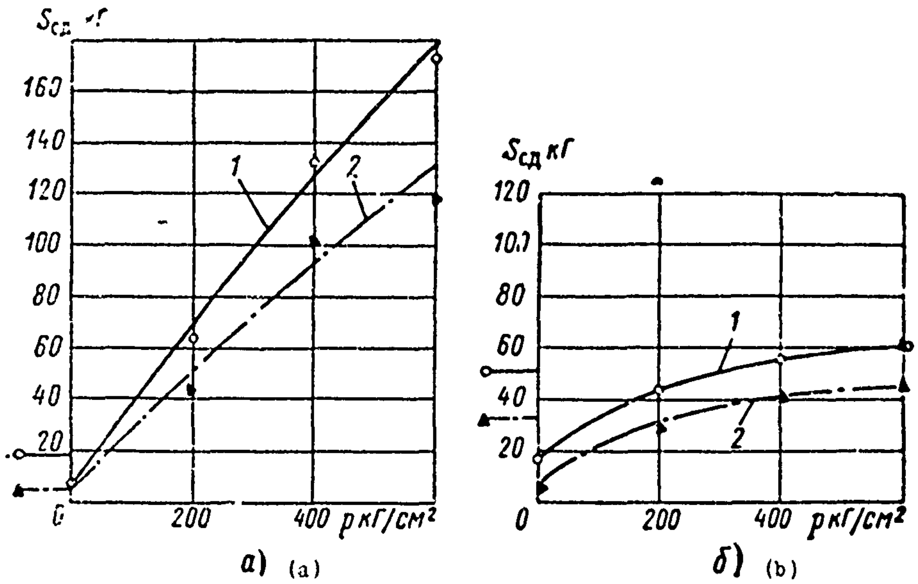

Fig. 3.12. Dependence of the initial shear forces $S_{сд}$ of the rings on the pressure (to the left of the axis of ordinates are plotted the values of the shear forces during movement without pressure after the storage of rings for three weeks): a - with moving rings; b - with stationary rings; 1 - rubber with a natural-rubber base SKF; 2 - rubber with a natural-rubber base SKMS-10.

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[1]This can be explained by the quality of the surface: thus, aluminum parts are always anodized after machining; the anodic film thus formed is porous, blistered, and has solid, sharp microunevenesses.

In automatic units in which long service life is required the rubber rings are protected by safety washers (Fig. 3.13) to reduce wear. These washers are made from teflon and have a thickness of 1.0-3 mm (depending on the pressure and the diameter). Protective washers should be installed with a pressure exceeding 150 at, and with large diameters (> 300 mm) - even at lower pressures.

Fig. 3.13. Protection of rubber rings by teflon washers.

Protective washers are installed with little clearance or even with interference. Such teflon rings protect the rubber ring from exclusion into clearance Δ, thereby reducing friction. With a one-sided pressure delivery the protective ring is installed only on one side.

Guard rings are usually made split, with an oblique cut (Fig. 3.14a). In certain designs unsplit protective rings are used (see Fig. 3.14b). Frequently, with little difference in the outer and inner diameters and with small thickness, a solid washer is inserted with the aid of an assembling ring, deforming the teflon (see Fig. 3.14c). A little while later, the dimensions of the ring are restored.

Rings made from teflon are not used as the basic seal.

3.3. BELLOWS

In pneumatic propellant valves of LPREs, especially in valves which handle aggressive and low-boiling oxidizers, bellows are widely employed to separate the controlling and the liquid cavities.

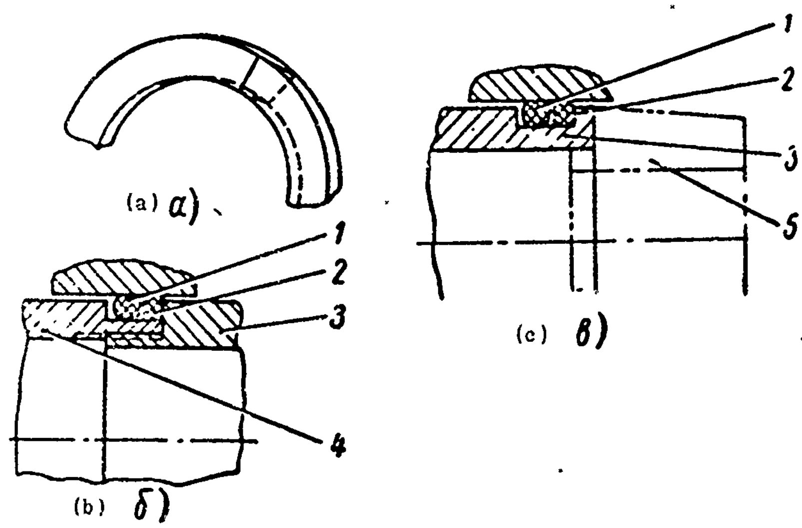

Fig. 3.14. Methods of installing safety washers: a - cut teflon ring (with oblique cut); b - releasable connection; c - installation of guard ring by deforming it; 1 - seal ring; 2 - safety washer; 3 - rod; 4 - bushing; 5 - assembling ring.

A bellows is a thin-walled corrugated elastic metallic box; by means of welding or soldering it is hermetically connected at one end with the moving system of the valve, and with the other end (through intermediate flange connections) - with the housing of the assembly. An example of such constructions can be the valve bellows depicted in Figs. 2.1, 2.5, and 2.10.

The fundamental and decisive advantage of a bellows seal over rubber seals and rings is the possibility of using it for handling corrosive and low-boiling liquids.

A bellows also has other important advantages over rubber seals, such as: stability of characteristics in connection with the absence of friction forces, which change depending on the temperature, amount of lubricant and so forth; moreover, a bellows seal ensures total permeability; the storage period of a bellows has little effect on the efficiency of the connection (the phenomenon of "aging" of the rubber is absent).

At the same time it should be noted that the manufacture of bellows is considerably more expensive and more laborious than the production of rubber seals. A design of an assembly with bellows is more complex and heavier than an assembly with seal rings. The service life of bellows is considerably less than the service life of rubber seals or rings (the destruction of bellows, especially single-layer, usually proceeds along the edges or corrugations next to them - cracks appear in them).

To increase the service life multi-layer bellows may be employed, but this complicates their production even more.

The service conditions of bellows in pneumatic-automatic assemblies of LPREs are considerably more difficult than the service conditions of bellows employed in instrument manufacture.

In the pneumatic propellant valves the bellows hold great pressure differentials, which change abruptly in value and direction (the excess pressure acts now on the inside, now on the outside of the bellows); the deformations are great and close to the maximum possible; bellows must operate with abrupt changes in temperature and under pressure and under conditions of vibrations, or, what is even more difficult, in vibration without pressure.

In designing the assembly it is essential to see to it that during vibration, caused by engine operation, the bellows cannot shift arbitrarily - that it cannot be compressed or pulled out. This is ordinarily ensured by the presence of pressure inside or outside the bellows. Sometimes the bellows is loaded only by the force of a spring, but this is less desirable, since the corrugation may vibrate. If the possibility of vibration of the bellows during engine operation is not eliminated, then its service life is sharply decreased.

The material employed for the manufacture of bellows must possess high plasticity. Bellows employed in assemblies of LPREs are ordinarily manufactured from stainless steel Kh18N10T; bellows manufactured from zinc alloys, low brass L80 or beryllium bronze of various brands are much more infrequently employed.

The structural special features of a bellows can be divided into:

a) single-layer and multi-layer;

b) manufactured from seamless tubes and tubes with seams;

c) reinforced by rings and unreinforced;

d) straight-through and with a bottom.

In the propellant valves of LPREs seamless-tube bellows, single-layer and multi-layer, are employed exclusively; in the majority of cases they are reinforced by outer or inner rings.

The construction of a bellows of a propellant valve is depicted in Fig. 2.2.

The fastening of the bellows to the fittings (to flanges, to the disk of the valves and so forth) is accomplished by means of welding or soldering.

Most frequently employed is welding, which must ensure high strength of the joint, its hermeticity and corrosion resistance. The use of welding predicates the use of materials identical or close in composition to the material of the bellows itself for the fittings of the bellows.

Soldering of bellows is employed in those cases (mainly for bellows manufactured from copper alloys), when heterogeneous materials are used for the fittings and bellows and when the connection is not very strongly loaded. Soldering may be employed

only in those cases where the temperature of operation of the bellows is significantly less than the melting temperature of the solder (approximately 130°C lower). Care should be taken so that the flux employed in soldering does not cause corrosion of the soldered joint or of the base metal of the parts. The dimensions of the bellows in the attachment diameters are shown in the appropriate industrial standards. Thus, steel single-layer bellows are standardized by norms MN428-60 - MN431-60[1].

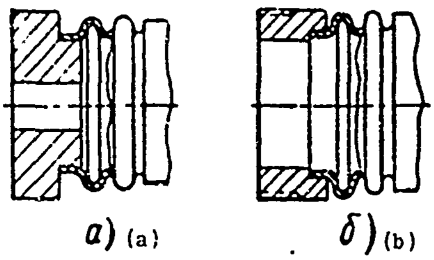

Fig. 3.15. Fastening of the bellows to the fittings: a - on the outer diameter of the fitting; b - on the inner diameter of the fitting.

The welding of the bellows to the fittings is ordinarily accomplished on the outer diameter of the fitting (Fig. 3.15a) in those cases where the outer pressure is the greatest calculated pressure acting on the bellows. Bellows which operate only under internal pressure, or such bellows, in which the internal pressure is most dangerous, are welded to the fittings on their inner diameter (see Fig. 3.15b).

The sizes of the bellows fastening diameters are ensured during corrugation, therefore the construction of the attachment is connected with the technology of manufacture of the bellows. The fastening diameter does not influence the value of the pressure, which must be supplied to the bellows for compression or elongation of it to a certain length In other words, the effective area

[1]See "Index of machine construction norms for 1966", published by the Committee of Standards, Measures and Measuring Instruments at the Soviet of Ministers of the USSR, 1966.

of the bellows (see below) does not depend on the mounting diameter of its ends.

In welding the connection of the bellows with the fittings the thickness of the walls of the joined parts usually do not correspond to one another (the width of the fittings must not be less than 1.5 mm), however at the present time it is possible even in this case to obtain a reliable connection.

The welding of the bellows connections to the fittings should be done by short-pulse seam welding (for the external seam), argon-arc or electric-arc welding. This produces accurate, dense and strong seams and reduces the probability of intergranular corrosion[1].

Such types of welding permit the automating of production.

With argon-arc welding the possibility of intergranular corrosion is reduced due to the quicker cooling as a result of blasting by the inert gas - argon. Moreover, the inert gas protects such component steels as chromium, titanium and others from burnout, which improves the quality of the seam and the near-weld section.

Cathode-ray welding is an even more advanced method, which avoids the intensive heating of the broad near-weld zone.

During ordinary welding with the use of a flux it is necessary to carefully check the welded seams and the regions ajoining it for intergranular corrosion.

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[1]Intergranular corrosion is corrosion on the edges of the grains of chrome-nickel steels of the austenitic class of type 18-8 (for example, 1Kh18N9, 1Kh18N9T and others), caused chiefly by slow cooling or heating within the temperature range of 450-850°C.

Figure 2.2 shows a design of a single-layer bellows operating with alternating action of the external and internal pressures.

The outer ring (casing) (position 4 in Fig. 2.2) serves to impart to the bellows rigidity during the action of the pressure inside the bellows - they prevent the buckling of the corrugations. Furthermore, the external rings serve to limit the magnitude of deformation of the corrugation - the compression of the bellows may occur only up to the moment of contact of the rings. In order to reduce weight (which is of great significance for the efficiency of the bellows under conditions of vibration, and especially in the case of vibration when there is no pressure), the external rings are manufactured from light metals or alloys, most frequently from aluminum.

Sometimes the external rings are made compound; in this case the ring consists of two half rings, onto which the upper (outer) strengthening ring 1 is pressed (Fig. 3-16). This design allows the installation of the external ring after the manufacture of the bellows connections, while with one-piece rings it is necessary to assemble the ring during the corrugation process (before the formation of the corrugations).

To avoid loss of axial stability by the bellows (especially for long bellows, located inside the assembly) guide bushings are sometimes used, which serve for centering the bellows along the outer rings or, where there are no rings, - directly on the external or internal diameters of the corrugations. These same bushings frequently serve as limiting supports during the compression of the bellows (see, for example, Figs. 2.5 and 2.8), taking the mentioned functions away from the external rings, which permits a simplification of the bellows, making the external rings narrower (i.e., taking the smaller dimension a, shown in Fig. 3.16). A somewhat greater longevity of the bellows unit is achieved under vibration conditions this means.

The internal rings impart stability to the bellows during the action of external pressure. These rings, are manufactured from wire OVS, are open circles (see Fig. 2.2), which permits them to be installed on the finished bellows. However, it is not convenient to install rings in the middle of a long bellows, and this circumstance to a certain degree limits the length of the bellows.

So that the sharp edges of the end of the ring cannot break the integrity of the bellows wall, bellows are frequently protected with special bands (coverings) made of fine soft metal, so-called "flag" 3 (see Fig. 2.2), slipped over one of the ends of the ring and fastened by soldering or welding. The other end of the ring is simply inserted inside the "flag". Due to the "flag" the longevity of the bellows is increased by many times. The clearance at the joint between the ends of the ring in the working state must be minimal - close to zero.

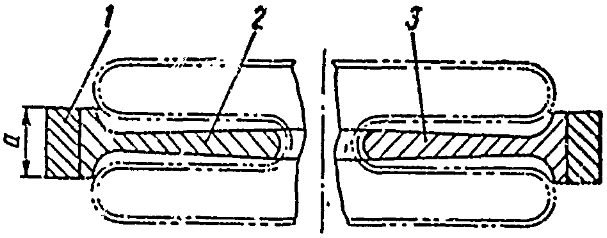

Fig. 3.16. A compound ring of a bellows: 1 - outer ring; 2, 3 - half rings.

The material of the "flag" is selected in such a way so as not to create an active voltaic couple, causing corrosion of one of the metals. (This refers, of course, also to the selection of the material of the ring, outer rings, flanges and so forth)

[illegible] service of the bellows (from the conditions of strength) [illegible] high degree of purity, must not have scratches, nicks, [illegible] cracks. This is ensured by the special technology of [illegible], correct organization of interfactory storage and [illegible]. Especially careful control is given to the surface

before drawing operations. Sometimes electropolishing of the surface of the bellows is employed.

For the use of bellows in propellant valves it is necessary to know more or less precisely the characteristics of the bellows.

The magnitude of the force which must be applied for compression or elongation of the bellows by 1 mm[1] is called the rigidity of the bellows k (by analogy with the rigidity of the spring):

$$k=\frac{P_1}{\Delta h},$$

where P_1 is the axial stress;
Δh is the displacement of the bellows.

The rigidity of the bellows is directly proportional to the modulus of elasticity of G of the material.

By the characteristics of the bellows we mean the dependence between the deformation of the bellows Δh (by compression or elongation) and the applied force necessary for this value. As can be seen from Fig. 3.17, the characteristic of a bellows is not a straight line; under great compression, when the corrugation of the bellows is deformed, the stress required for compression of the bellows significantly increases; the force also increases for expansion of the bellows during considerable elongation, which is also related to the deformation of the corrugations. For the regions of small deformation of the bellows f_0, where the deforma-

[1]The rigidity value is influenced by the angle α of the corrugation profile and also by the radii of rounding of corrugations R_1 and R_2.

tion does not cause considerable changes in the geometry of the corrugations, the rigidity of the bellows is approximately constant, and in this segment the characteristic of the bellows is linear[1].

The rigidity of the bellows after its manufacture is not a stable value. To impart stability to the bellows it is swaged, repeatedly reduced by pressure, i.e., it is subjected to operations similar to the compression of springs. However, even after this it cannot be assumed that complete stability of the characteristics of the bellows has been achieved during considerable deformations of it. The bellows possess hysteresis, i.e., the forces necessary for the deformation of the bellows in the forward and opposite directions are nonuniform (Fig. 3.18).

For the bellows of propellant valves, in contrast to bellows used in instrument manufacture, the requirement for stability of the characteristics is not decisive or basic. The variation in rigidity of a bellows during fluctuations of the surrounding temperature within a range of $\pm 50°C$ is immeasurably less than the change in the friction force when using cup packings.

To determine the force which acts on the bellows when pressure is supplied inside the bellows or outside it, the concept of the effective area of the bellows s introduced.

By the effective area $F_э$ we mean the value of the area, on which under the effect of the excess pressure p is obtained the elongation of the bellows Δh, equal to the elongation of the bellows under the effect of the concentrated force P_1.

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[1]Characteristics of bellows manufactured from steel 36NKhTYu (EI702) are cited in industrial standards.

From this determination it follows that:

$$pF_э = k_1\Delta h; \quad \Delta h = \frac{pF_э}{k_1} = \frac{P_1}{k},$$

where k_1 is the rigidity of the bellows during deformation of the bellows by a value Δh under the effect of pressure p.

The characteristic of the bellows almost does not depend on the form of the applied force: both under the loading of the bellows by an axial concentrated force, as well as under the effect of excess pressure on it, the obtained characteristic of the bellows is approximately identical. Therefore values k and k_1 of the rigidity of the bellows almost do not depend on the loading conditions[1], i.e., $k_1 \approx k$.

Then

$$F_э = \frac{P_1}{p}.$$

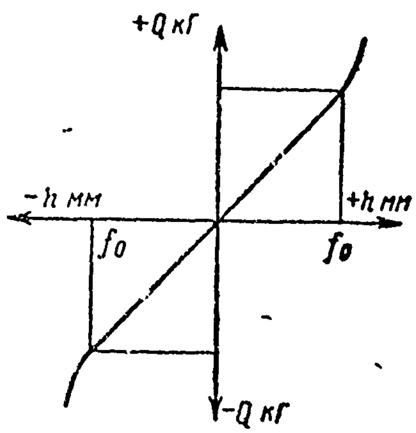

Fig. 3.17. Characteristic of the bellows.

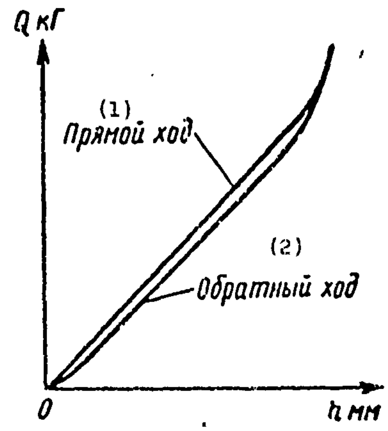

Fig. 3.18. Hysteresis of the characteristic of the bellows. KEY: (1) Forward travel; (2) Reverse travel.

[1]Actually, loading at the walls of the material of the bellows is distributed differently, depending on whether loading by a concentrated force or by gas (liquid) pressure, distributed over the entire area of the bellows, is accomplished.

Thus, the effective area is an arbitrary value and not exactly constant. With an increase in the value Δh, as a result of the divergence in the values of the coefficients k_1 and k, value $F_э$ varies.

With small movements (within the limits of the linear characteristic of the bellows) the effective area is numerically equal to the area of a circle with a diameter equal to the half-sum of the outer and inner diameters of the bellows:

$$F_э=\frac{\pi D_{cp}^2}{4},$$

where

$$D_{cp}=\frac{1}{2}\cdot(D_{вн}+D_н).$$

According to the data obtained from the experiments conducted the effective area of the bellows depicted in Fig. 2.2 is actually 2-3% greater than the area determined according to formula $\pi D_{cp}^2/4$ and with an increase in the compression of the bellows it increases slightly, which can be explained by the change in the geometry of the bellows corrugations.

The magnitude of compression (elongation) of the bellows under the effect of the applied force with all other conditions being equal is directly proportional to the number of corrugations:

$$\frac{\Delta h_1}{\Delta h_2}=\frac{n_1}{n_2},$$

where n_1 and n_2 are the number of corrugations of two identical bellows;

Δh_1 and Δh_2 are the elongations of bellows under equal loading.

This proportionality is valid with a sufficiently large number of corrugations, which excludes the influence of extreme corrugations, adjoining the frame (to the site of welding of the bellows to the flanges), the rigidity of which may differ from the rigidity of the average corrugations.

The deformation of the bellows is approximately inversely proportional to the cube of the thickness of the wall of the corrugation s:

$$\frac{\Delta h_1}{\Delta h_2}=\frac{s_2^3}{s_1^3},$$

in other words, the rigidity of the bellows is directly proportional to the cube of the thickness of the corrugation.

This relationship is valid only with small radii of curvature of not-too-compressed bellows. With thin walls the index of the degree with thickness s is somewhat reduced (from the threes to 2.7-2.8).

The deformation of the bellows depends also on the height of the corrugation (the difference between the outer and inner diameters of the bellows).

The following dependence was obtained experimentally:

$$\frac{\Delta h_1}{\Delta h_2}=\left(\frac{D_{\text{н}_1}}{D_{\text{вн}_1}}\right)^2:\left(\frac{D_{\text{н}_2}}{D_{\text{вн}_2}}\right)^2.$$

The experimental data testified to the fact that an increase in the radius of curvature of the bellows corrugations promotes an increase in their rigidity.

Ordinarily the deflection (compression) of each corrugation does not exceed values of 0.025 D_H for bellows of small diameters (up to an outer diameter value of $D_H \approx 40$ mm), and 0.015 D_H for large diameters.

All the above-cited formulas and remarks refer to one-layer bellows.

When using bellows in repeat-action assemblies, especially in test bench assemblies, guaranteed service life is of great significance.

The service life of a bellows seal is affected by a number of factors, enumerated below.

The value of deformation (elongation or compression), taking place on one corrugation. The lower the deformation, the longer the service life, and therefore if necessary for an increase in the bellows service life the number of corrugations should be increased. This is one of the most effective methods.

The neutral position of the bellows must be selected, such that the deformation is distributed symmetrically; in one of the extreme positions the bellows is extended relative to the neutral position, and in the other it is compressed. However, more often attempts are made to select the bellows dimensions in such a way that with maximum active internal pressure the bellows is in the neutral position, while with reduced pressure it is compressed.

The thickness of the bellows wall s. An increase in the thickness s of the wall reduces the operating service life. The value s should be selected as the minimum possible from the conditions of strength and stability of the bellows.

The character of loading. A smooth pressure supply significantly increases the service life in comparison with abrupt loading. The bellows operates better with external pressure than with internal, - with external loading it is capable of maintaining a pressure of 1.5-2 times higher than with internal; with external pressure the axial stability of the bellows is better maintained.

The presence of inner and outer rings. The presence of rings provides long service life for the bellows by reducing the stresses acting in the material.

The pressure magnitude. A reduction in the pressure with all other conditions being equal reduces the stresses in the bellows, as a result of which its service life is increased.

The number of layers of the bellows. A fundamental means of increasing the service life (just as the increase in the number of corrugations) is the application of a two- or multi-layer bellows.

Thus, a three-layer bellows with a thickness of each wall of 0.15 mm ensures a service life of 3340-7360 triggerings, while the guaranteed service life for a single-layer bellows with a wall thickness of 0.3 mm for the same conditions amounts to only 600 operations.

Multi-layer bellows are employed when large differentials in pressure act on the bellows. The multi-layer bellows operates according to the principle of a multi-band spring and reduces the stresses in the bellows wall. With one and the same deflection a multi-layer bellows has a much greater safety factor, than a single-layer with a wall thickness equal to the total thickness of the walls of all the layers of a multi-layer bellows.

The manufacture of multi-layer bellows is more complex than single-layer. When using multi-layer bellows inter-layer nonhermeticity, i.e., nonhermeticity of one of the layers, is a danger to be avoided. Such leakage can arise during welding of the layers to one another; sometimes during operation in one of the layers a minute crack or a pin hole can develop in welding. Such interlayer leakage leads to the fact that the compressed gas under pressure enters the clearance between the layers, but with the release of pressure remains there (or goes out very slowly). Then the bellows remains very rigid, as a result of which the normal functioning of the valve breaks down. Therefore, during manufacture of two-layer and multi-layer bellows after welding the layers to one another care should be taken to check the bellows for interlayer nonhermeticity.

The simplest method is the following: the bellows is placed in some kind of container, into which a working gas pressure (100-200 at) is introduced and the bellows is maintained in this state for 5-10 minutes (there is pressure outside and inside the bellows, i.e., the bellows does not take any kind of loads). Then the pressure is released, and the bellows is quickly immersed into a bath with water or, what is more desirable, with gasoline, so that the liquid barely covers the bellows. If there is interlayer nonhermeticity, at the spot where the defect is, small bubbles of gas penetrating through the layers will begin to emerge (they are especially clearly visible in gasoline). Here is it also possible to determine the location of the leakage - the inner or outer layer, or the welding area.

CHAPTER 4

THE SEALS OF THE LOCKING MECHANISMS OF PNEUMATIC PROPELLANT VALVES (THE END SEALS)

In pneumatic assemblies the hermeticity of the locking mechanisms is most frequently ensured by the seating of the valve disk on a seat, built into the valve housing. In the disk there is fastened a ring, made from a relatively soft packing material - rubber, capron, teflon and so forth[1]. The pressure in the control cavity or the spring force creates the required force to press the valve disk onto the seat.

Rubber is used for end seals, operating in the temperature range of $\pm 50°C$ in air, alcohol, kerosene, hydrogen peroxide and other, comparatively low-corrosive, propellants.

Rubber rings are fastened in the valves in the following manner. Onto the valve head is fastened a ring made of crude rubber, after which the rubber is vulcanized under a press; the fastening of the rubber to the metal is accomplished by Layconat glue. The shape of the groove in the head of the part is shown in Fig. 4.1a. To ensure reliable adhesion the groove under the

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[1]The sealing in explosive valves is usually accomplished by wedging one part into another. What is said below refers only to the case where soft seals are employed, which is sometimes the case in explosive check valves.

rubber should be machined to a finish of ∇3 and then sand blasted to obtain surface roughness. The more roughnesses the surface has, the more reliable will the adhesion of the rubber be. Layconat glue produces the best adhesion with carbon steels for aluminum alloys.

However Layconat cement is not resistant to certain propellants. Therefore, in handling such propellants to ensure reliability of adhesion of the rubber, a special shape of groove is employed (see Fig. 4.1b).

The undercuts which are in the groove are filled during compression, and after vulcanization the rubber is capable of resisting the forces which pull it from the channel.

The form of the billet of a rubber-metal part after vulcanization is shown in Fig. 4.1c, on the central rod there is centered a punch, which compresses the rubber during vulcanization. On surfaces A and B there can be rolls of rubber. After vulcanization the surface of the rubber is mechanically worked.

Other methods of fastening the rubber - lapping, press fitting and so forth - result in less strength and therefore they are not widely used.

Many types of propellant cause swelling of the rubber. Thus, when the rubber seal in the valve in the assembly shown in Fig. 2.10 was held in the propellant for 24 hours, it swelled (it increased in height above the metal by a value of up to 0.3 mm). Such swelling of the rubber, which does not affect the hermeticity of the seal, causes an increase in the hydraulic resistance of the valve during operation in prestage mode.

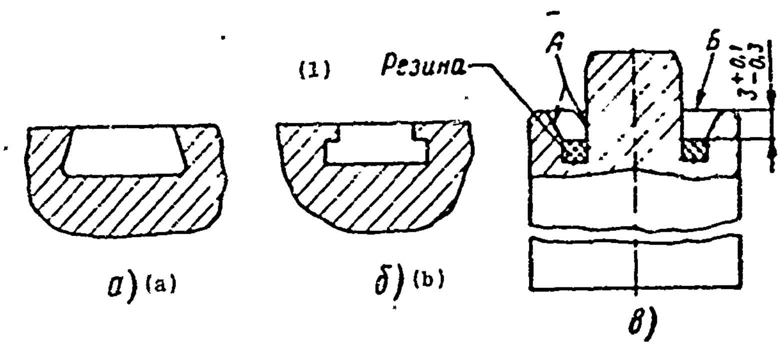

Fig. 4.1. Shape of the groove of a rubber seal (in the finished part): a - with reliable adhesion of the rubber; b - with a cement which is unstable with respect to the propellant; c - view of a billet of a rubber-metal part after vulcanization.
KEY: (1) Rubber.

For normally closed and normally covered valves, where the seat is constantly pressured by the springs against the seat, the phenomenon of "adhesion" of the rubber to the seat should be taken into account. The "adhesion" value depends on the temperature; maximum adhesion lies in the temperature zone which exceeds by 10-20°C the temperature of vitrification of the rubber; with a further reduction in temperature the "adhesion" is sharply reduced. The value of this "adhesion" increases with the passage of time; it is determined chiefly by the form of the rubber. The maximum value of the adhesion force may reach 0.5 kgf for 1 mm of length of the perimeter of the seat.

As a result of the phenomenon of "adhesion" the force required to open the valve varies, i.e., its operating stability deteriorates.

In connection with the appearance of increasingly more aggressive propellants the region of application of rubber as a sealing material has decreased.

Teflon has received wide application as an end seal of pneumatic valves. Teflon is a polymer of tetrafluoroethylene. The distribution and use of this material is explained by its exclusive chemical stability, which exceeds the stability of all now-known sealing materials. It is, practically speaking, stable under the action of the overwhelming majority of fuels and oxidizers. Only under the effect of molten alkali metals and fluorine does it decompose, according to reference data.

Teflon is pressed into billets in the heated state beneath a press. The groove for the teflon seal has the shape of a dovetail. The inner surface of the groove is machined to a finish of ∇3-∇4. The pressed teflon insert has a diameter equal to the average diameter of the groove (Fig. 4.2). The height of the teflon billet amounts to 5-7.5 mm.

The teflon billet temperature during pressing must be 140-150°C. The specific pressure during pressing amounts to 800 kgf/cm^2. Under the effect of the pressure in the hot state the material fills the channel intended for it. The teflon does not manage to stick to the groove, since because of its chemical inertness it reliably does not bond with one of the cements known thus far.

For better adhesion of the teflon in the groove several (from two to eight) special bore holes 1 and 2 (see Fig. 4.2) Ø0.8-1.2 mm are frequently made in the billet; through these holes during the pressing of the seal air (hole 1) exits and the teflon is extruded (hole 2).

The pressing technology and quality is the decisive influencing factor on the working efficiency of the seal. With poor pressing leakage of air is possible around the seal along the perimeter of the groove (Fig. 4.3). This ordinarily takes place at low

temperatures, when as a result of a large coefficient of linear expansion of the teflon between it and the walls of the groove a clearance forms.

Under certain operating conditions, for example, while checking the hermeticity of the seals of the assembly with compressed air, a similar phenomenon may lead to the pressing out of the teflon from its seat. With a closed valve (see Fig. 2.3) the high pressure air, located in the liquid cavity[1], slowly penetrates beneath the teflon, which is used as the seal of part 2; with the abrupt release of pressure from the cavity of the valve the air cannot escape quickly from under the teflon; a pressure differential is produced on the seal, which leads to the extrusion.

Since during operation with a liquid such cases are practically impossible, to avoid this defect while checking, the slow release of the air from the valve cavity is provided for.

Moreover, with the same aim in mind, frequently right after the pressing of the teflon a bore hole 1 is made (on the outlet side of the assembly) (see Fig. 4.3) to remove air which enters when pressure is supplied for the sealing.

After the pressing of the seal one or two parts from the batch should be cut open, in order to determine the quality of the press fit and the degree of filling of the channel with teflon.

It is difficult to achieve total hermeticity at low temperature when handling compressed air or another gas, especially under high-pressure, using a seal made from teflon. But, because of the absence of brittleness and the high wear resistance, such a seal, as a rule, does not break down and does not wear out.

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[1]While checking the hermeticity of seal 15 (see Fig. 2.3).

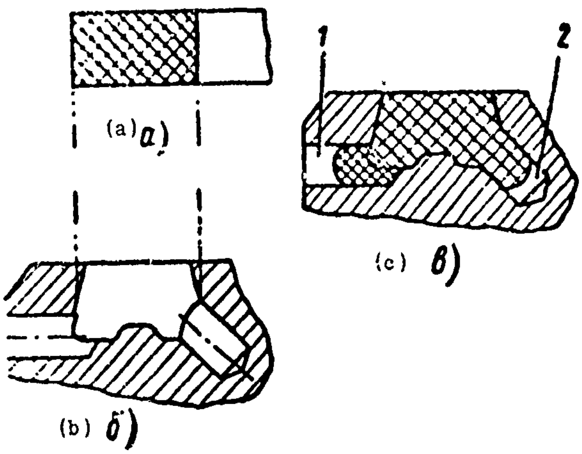

Fig. 4.2. Teflon seal: a - teflon billet; b - groove for the teflon; c - after pressing; 1 - channel for escape of air when pressing; 2 - bore hole for attachment of teflon.

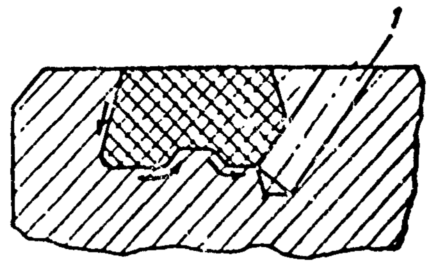

Fig. 4.3. Path of the air around the seal (shown by arrows): 1 - passage for discharge of air (bored after the teflon is pressed in).

Other, chemically stable common materials are also used as seals - polytrifluorochloroethylene (PTFCE). It is considerably harder than teflon, but somewhat resembles it in its chemical resistance[1]. PTFCE can operate with higher specific pressures than teflon.

In approximately the same manner as teflon (but in the cold state), polycaprolactam (capron) is pressed. It is rarely used, and only for noncorrosive products. Sometimes capron is charged into the valve.

The hermeticity of the end seals of locking mechanisms is determined by the factors cited below.

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[1]In practice, PTFCE can be used for the same types of propellants, for which teflon is used.

4.1. PHYSICO-MECHANICAL AND CHEMICAL PROPERTIES OF A SEALING MATERIAL

The physico-mechanical properties of a material determine its possibility for application for a given unit, considering the working conditions - the service life, pressure, temperature conditions, and required hermeticity. In view of the special properties of propellants the majority of ordinary sealing materials are not suitable for use in the propellant valves of LPREs.

For operation in the end seals with less corrosive propellants special rubbers are used: for alcohol, hydrogen peroxide (and sometimes for liquid oxygen) - rubber based on butadiene-styrene natural rubber SKS-30; for petroleum-based fuels - rubber based on butadiene-nitrile natural rubber of brands SKN-26 or SKN-26 + SKN-18. For inert gases and air there is rubber based on natural rubber SKN-40 + SKN-18.

The softer the rubber, the lower is the required specific pressure in the seal to ensure hermeticity at low temperatures. However, with soft types of rubber there exists the danger of breakdown of the sealing material with repeated operations.

The hermeticity of the seal should be checked at the lowest operating temperatures, since this corresponds to the most rigorous working conditions.

The stability of the rubber in the propellant, on the other hand, should be checked at the maximum possible operating temperatures, when the reacting capacity of the propellant is increased.

Attempts have been made to use rubber for end seals of normally open valves of the balance type, which handle liquid oxygen. To ensure the hermeticity a high seat, shown in Fig. 4.4a,

was employed. The presence of the cylindrical section of the seat profile (segment a) led to the wedging of the seat in the rubber after oxygen was supplied (with the valve closed) as a result of the difference in the coefficients of linear expansion of the rubber and metal. Since the valve opens under the action of a relatively weak spring force, such a phenomenon led to a delay in the opening of the valve. The wedging was aggravated, if the surface a was made[1] with a slight inverse conicity (with the tip at the base of the seat, see Fig. 4.4b).

Conical seats of ordinary shape (see Fig. 4.5) work satisfactorily with a seal made from rubber. However, with respect to the permissible operating lifetime the rubber seal was inferior to the teflon one, as a result of which the rubber was replaced by teflon.

Teflon has great advantages over rubber with respect to chemical stability. Teflon may be used in assemblies of LPREs and temperatures from -196°C to +250°C. Although thin bands (films) of teflon may burn, yet in practice, with a sufficiently thick layer (3-4 mm), teflon does not burn. Teflon possesses a very low coefficient of friction and dielectric qualities. A disadvantage of it (just like PTFCE) is its high coefficient of linear expansion α, which differs from the value α of all employed structural materials[2]. As a result of this, a teflon seal works poorly with periodic temperature changes.

Teflon possesses creep - a property, characterized by the fact that with the passage of time under the action of a constant load the material is continuously and slowly deformed. The stress at which creep sets in, is small for teflon, and its value decreases

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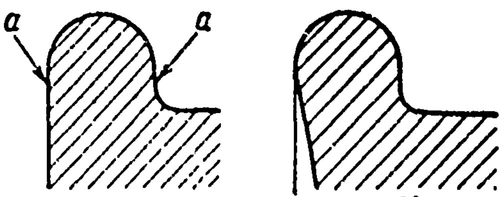
[1]As a result of errors in the manufacture of the special cutting tool.

[2]See Table 4.2.

with the rise in temperature. Thus, the stress for the start of creep varies from 1.42 kgf/mm^2 at t = 25°C to 0.47 kgf/mm^2 at t = 150°C. Residual deformations in teflon are very considerable. Thus, noticeable residual deformation in teflon at t = 20°C begins with a pressure of 0.3-0.5 kgf/mm^2, and with a pressure of 2-2.5 kgf/mm^2 the material flows. Therefore, after the impression of the seat into the teflon seal a depression from the seat is immediately obtained in the seal, and the depression is so much the deeper, the higher is the seat and the greater the specific pressure.



(a) а) (b) б)

Fig. 4.4. The seat of an oxygen valve: a - with a cylindrical section of the profile; b - with conicity.

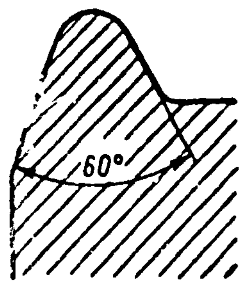


Fig. 4.5. Ordinary profile of a seat of an oxygen valve for a rubber seal.

The residual deformation with uniform specific pressure increases with a rise in temperature. In view of this, the hermeticity of the end seal, operating at elevated temperatures or situated for a long period under the action of a considerable load at high temperature, decreases if the seal is subsequently operated under low temperature conditions. This is explained by the fact that at low temperature the metal of the valve rests against the housing, while the seat is barely in contact with the teflon, on which the deep depression was previously formed. When operating with a seal made from teflon it is not desirable that, during the operation of the assembly, turning of the valve relative to the seat might take place, since a loss in hermeticity

might occur due to the residual deformation of the teflon. When operating with a rubber seal as a result of the elasticity of the rubber the transition from high temperatures to low proceeds without special difficulties.

It is recommended that the specific pressure which acts on a seal made from teflon during storage not exceed 30 kgf/cm^2, and for a seal made from PTFCE - that it not exceed 100 kgf/cm^2. For comparison let us say that for capron the permissible specific pressure during storage is 300 kgf/cm^2.

In view of the indicated properties of fluoroplastic it is employed chiefly in normally open valves. PTFCE, which has higher strength, is used under higher pressures than is teflon. However, in connection with the appearance of brittleness when heated PTFCE is used at operating temperatures which do not exceed +70°C.

Data on the physico-mechanical properties of plastics are shown in Table 4.1.

4.1.1. The Load Value on the Sealing Material

The hermeticity of a joint depends on the value of the specific pressure acting on the seal from the side of the seat.

To analyze the effect of the type of seat - its profile, height, construction - on the hermeticity, the concept of linear load ρ is used. The linear load ρ (in kgf/mm) is the force necessary per unit of length of the circumference of the seat (with respect to the average diameter D_{cp}):

$$Q = \frac{Q}{\pi D_{cp}},$$

where Q is the load in kgf.

Table 4.1. Physico-mechanical properties of plastic seal materials.

(2) Свойства	(1) Материал		
	(3) Фторопласт-4	(3) Фторопласт-3	(4) Капрон
(5) Температура плавления, °С	327 (6) Для кристаллитов	208÷210	210÷218
(7) Плотность, $г/см^3$	2,35 (6) Для кристаллитов	2,09÷2,16	1,1÷1,13
(8) Удельная ударная вязкость, $кГ \cdot см/см^2$	100	20÷160	100÷130
(9) Предел прочности, $кГ/см^2$			
(10) а) при растяжении	140÷250	350÷400	550÷700
	(11) для незакаленных образцов		
(12) б) при статическом изгибе	110÷140	600÷800	900÷1000
(13) в) при сжатии	—	200÷570	850÷1000
(14) Твердость по Бринелю, $кГ/мм^2$	3÷4	10÷13	10÷12
(15) Относительное удлинение при разрыве, %	250÷500	20÷40	100÷150
(16) Остаточное удлинение, %	250÷350	—	—
(17) Коэффициент линейного расширения	$(8÷21)\cdot10^{-5}$	$(6÷12)\cdot10^{-5}$	$(8÷10)\cdot10^{-5}$
(18) Интервал температур эксплуатации, °С	+260÷−269	+125÷−195	+50÷−50

KEY: (1) Material; (2) Properties; (3) Teflon; (4) Capron; (5) Melting temperature, °C; (6) For crystallites; (7) Density, g/cm^3; (8) Specific impact strength, $kgf \cdot cm/cm^2$; (9) Ultimate strength kgf/cm^2; (10) during elongation; (11) for untempered samples; (12) during static bending; (13) during compression; (14) Brinell hardness, kgf/mm^2; (15) Elongation per unit length at rupture, %; (16) Permanent elongation, %; (17) Coefficient of linear expansion; (18) Operating temperature interval, °C.

To determine the loading one should take into consideration the force of the springs, the force from the pressure of the working medium which loads or unloads the seal (see Fig. 4.13 below).

Since the value of ρ depends neither on the width, nor on the height of the seat, it does not reflect the true value of the stresses in the seal; however, it makes it possible to compare seals which are of the same design with one another and to evaluate the efficiency of seats of various types.

It is understood that the higher is ρ, the higher the hermeticity will be with all other conditions being equal. This is valid (just as is everything said below) until the metal of the valve reaches the stop in the seat (Fig. 4.6), i.e., until deformation of the material of the seal occurs. After the valve stops in the seat, further loading of the valve is absorbed by the metal of the valve and has little effect on the hermeticity of the joint.

The greater ρ is, the shorter will the service life of the rubber seal be with all other conditions being equal, since excessively great loads can lead to the breakdown of the integrity of the rubber.

The lower the operating temperature, the higher the value of ρ must be, so as to ensure the required degree of hermeticity. Therefore, the minimum essential value of ρ, adequate to create hermeticity (ρ_{min}), is determined at the lowest operating temperature. The value of ρ_{min} guaranteeing hermeticity is determined by the height and profile of the seat. With aging of the rubber the value of the required linear load ρ increases.

The value of the required minimum linear load ρ_{min} was determined experimentally for rubber based on natural rubber SKN-40 + SKN-18 depending on the height, type and profile of the seat.

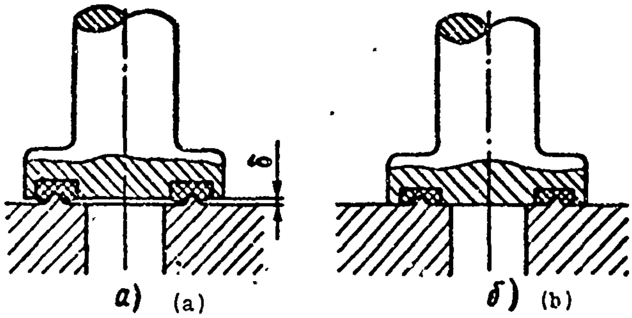


Fig. 4.6. Diagram of a seal of a locking mechanism: a - metal of the valve does not rest against the seat (clearance δ exists); b - the end face of the valve has reached the stop in the seat.

To ensure hermeticity of a rubber seal the value ρ at temperatures 15-25°C must not exceed 25 kgf/mm and must not be lower than 0.2 kgf/mm. Thus, with a seat[1] with an angle α = 90° and a height h = 0.6 mm the required value of ρ when working in air, with an excess air pressure p = 10 at, at a temperature of +20°C is 0.1 kgf/mm, and at a temperature of -50°C ρ = 0.7 kgf/mm.

The effect of the sealed pressure is determined by the following relationship (Fig. 4.7)

$$\rho_{min_p} = \rho_{min_0} + \beta p,$$

where ρ_{min_0} is the minimum linear load at low excess pressure (~10 at);

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[1]See Fig. 4.8, type I.

β is the proportionality factor;

ρ_{min_p} is the minimum linear load at pressure p.

The values β and ρ_{min_o} depend on the properties of the rubber and the seat design. The lower the seat, the higher the value of ρ_{min} must be. With sharper seats (with greater specific pressures) the values of ρ_{min} decrease.

For normally closed valves (the seals of which during storage are loaded by compressed spring pressures) attempts should be made to reduce the value of the linear load during storage, since it leads to residual deformations in the material of the seal. Thus, for seats, the profile of which is shown in Fig. 4.8 (type I) and the data for which are shown in Table 4.2, the value of the linear load during storage is limited by the value 0.1 kgf/mm. For normally covered valves the maximum value of linear load during storage may be increased up to 0.4 kgf/mm (since during operation when pressure is supplied the specific pressure will be increased relative to its value which existed during storage).

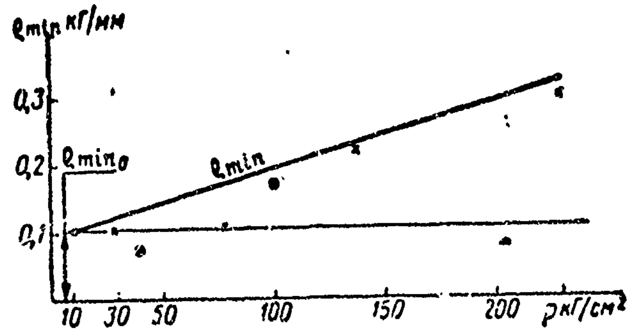

Fig. 4.7. Dependence of the minimum permissible linear load ρ on pressure.

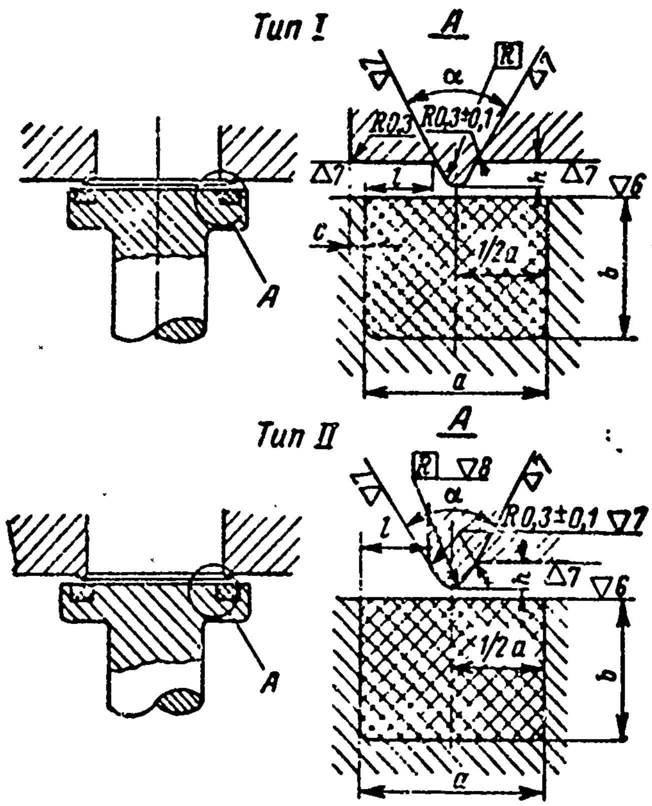

Fig. 4.8. Seats for rubber seals: type I - closed seat; type II - open seat.
KEY: (1) Type.

Table 4.3 presents new data characterizing the parameters of locking mechanisms with rubber seals; these locking mechanisms exhibited satisfactory working capacity in the temperature range of ±50°C.

Table 4.4 shows data on the specific pressure and the linear load for sealing units employing teflon. The cited designs of these units were successfully employed and produced satisfactory results with respect to hermeticity. The types of seats mentioned in Table 4.4 are shown in Fig. 4.9.

Table 4.2 Recommended dimensions of a seat of type I (see Fig. 4.8) for rubber based on natural rubber SKN-40 + SKN-18.

Working pressure up to 230 at. Compressed medium: air, nitrogen, inert gases.

Linear load kg/mm		Number of operations	Dimensions of grooves, mm		Dimensions of seat		
ρ_{min}	ρ_{max}		a	b	R mm	h mm	a degrees
0.7	25	500-100	4	2.5	0.3	0.6	90
0.8	25	800-400	5	2.5	0.3	0.6	120
1.5	25	1000-600	4	2.5	0.3	0.4	120

Remarks. 1. A long service life (a great number of operations) refers to the lower value of ρ.
2. The dimension l must be greater than 1 mm; the dimensions ≈ 0.6 mm (see Fig. 4.8).

4.1.2. Width and Depth of the Groove for the Seal

The working capacity of a seal depends on the volume of material of the seal, on which the deformation caused by the loading of the seat is distributed.

The problem of the essential dimensions of the grooves for a seal has been most completely studied for rubber seals. The failure of a rubber seal with the increase in the number of operations occurs in different forms, depending on the temperature.

Table 4.3. Recommended types and dimensions of seats for rubber based on natural rubber SKS-30 (see Fig. 4.8).

Working pressure up to 100 at. Compressed medium: air, nitrogen, water, alcohol.

Linear load Q kgf/mm	Dimensions of the groove, mm		Dimensions of the seats					
			closed seat type I			open seat type II		
	a	b	α degrees	R мм	h мм	α degrees	R мм	h мм
up to 2	3	2,5	60	0,2	0,4	60	0,2	0,4
	4	2,5	90	0,2	0,6	90	0,2	0,6
	4	2,5	90	0,2	0,4	90	0,2	0,4
2÷10	4	2,5	90	0,3	0,6	90	0,3	0,6
	5	3	120	0,1	0,4	120	0,1	0,4
10÷20	5	3	120	0,1	0,6	120	0,2	0,6
	6	4	120	0,1	0,4	120	0,1	0,4
	6	4	120	0,2	0,6	120	0,2	0,6
	6	4	120	0,3	0,9	120	0,3	0,9

Table 4.4. The use of teflon in the locking mechanisms of propellant valves.

Type of profile of the seat (see Fig. 4.9)	Average diameter of the seal	Linear load, Q		Specific pressure		Dimensions of the groove		
		in storage	in operation	in storage	in operation	b	h	α
	D_{cp} мм	кГ/мм	кГ/мм	кГ/мм²	кГ/мм²	мм	мм	degrees
I	4,7	0,0338	0,85	0,0564	1,42	Valve manufacture entirely from teflon		
I	5	0,098	1,16	0,163	1,93			
I	5	0,0395	1,02	0,066	1,7			
I	5	0,0395	0,077	0,066	0,128			
I	5,04	0,212	6,51	0,354	10,86			
I	15	0,490	3,19	0,815	5,31			
I	15	0	3,21	0	5,35	3	2,5	15
I	18,5	0,196	1,05	0,327	1,75	2,5	2,5	15
I	20	0,35	2,04	0,922	3,4	3	2,5	15
I	20	0,535	2,04	0,891	3,4	3	2,5	15
I	28	0,362	6,21	0,544	10,3	3	2,5	15
I	29	0,361	5,34	0,6	8,9	3	2,5	15
II (l=1)	29	0,0044	4,35	0,0044	4,35	—	—	—
II (l=1)	41,5	0,174	3,1	0,174	3,1	3	2,5	15
II (l=1)	42	0	4,21	0	4,21	3	2,5	15
II (l=1)	42	0	5,14	0	5,14	3	2,5	15
II (l=1,2)	26	0,274	4,35	0,228	3,63	5	2,5	15
II (l=1,2)	29,2	0,275	0,57	0,23	0,475	3	2,5	15

Table 4.4. (Continued)

Type of profile of the seat (see Fig. 4.9)	Average diameter of the seal	Linear load, q		Specific pressure		Dimensions of the groove		
		in storage	in operation	in storage	in operation	b	h	α
	D_{cp} мм	кГ/мм	кГ/мм	кГ/мм²	кГ/мм²	мм	мм	degrees
II (l=1,2)	53,2	0,353	4,27	0,294	3,56	4,5	3,5	15
II (l=1,5)	91,5	0,91	0,91	0,606	0,606	4	2,5	15
II (l=2)	81	1,36	1,36	0,68	0,68	4	2,5	15
II (l=2,5)	117,5	2,29	15,4	0,96	6,16	7	3	20
II (l=2,5)	127,5	2,14	12,15	0,856	4,86	7	3,75	20
II (l=2,5)	142,5	1,9	12,6	0,76	5,04	7	3	15
II (l=3,6)	75,6	0,97	13,8	0,27	3,84	1,5	3,75	15
III	115,8	0	5,9	0	2,95	4,5	3,5	15
III	116	0	5,85	0	2,925	4,5	3,5	15
III	118	0	5,74	0	2,87	3	2,5	15
III	135,8	0	5,9	0	2,95	4,5	3,5	15
III	136	0	5,85	0	2,925	4,5	3,5	15
IV	13,5	0,016	0,016	0,032	0,032	3	2	—
IV	20	0,0237	3,53	0,474	7,06	3	2,5	15
V	56	0,864	0,864	1,08	1,08	3	2,5	15
VI	10,8	0	2,87	0	7,16	3	2;5	15

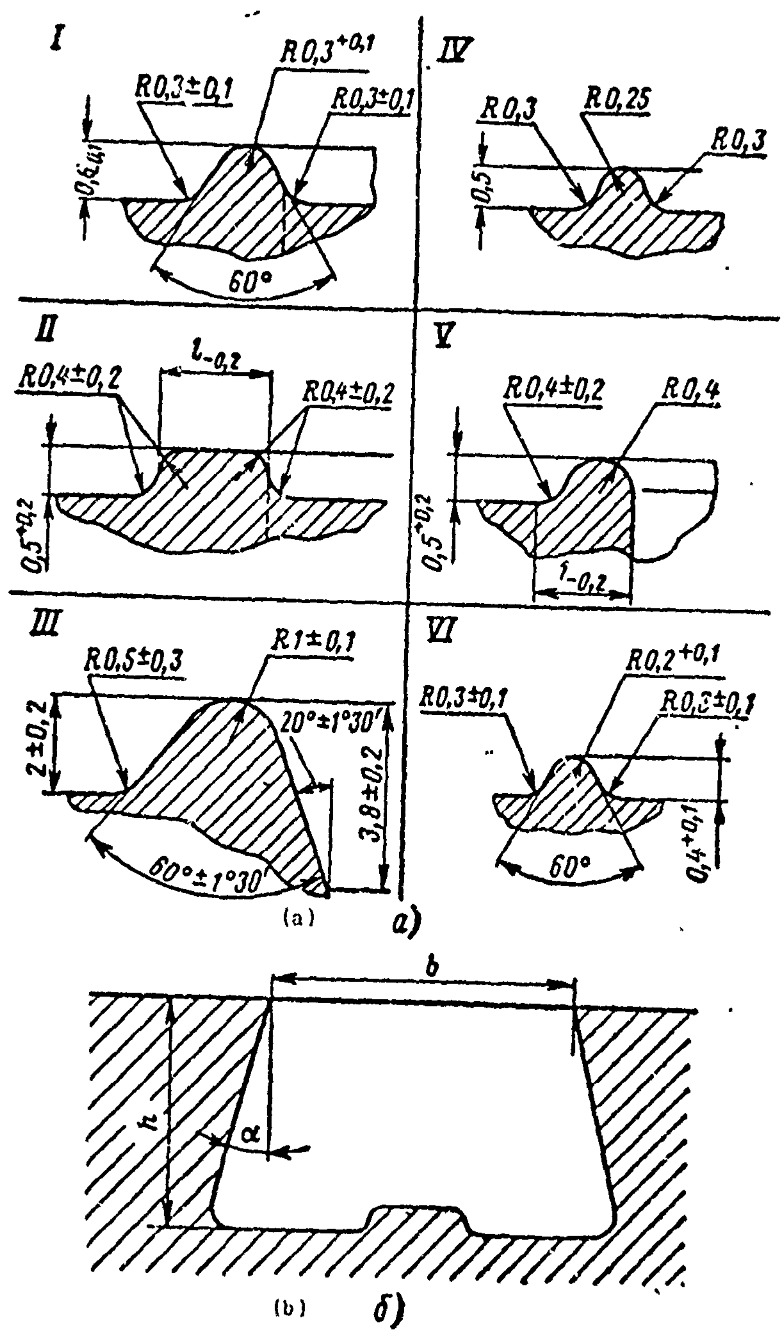

Fig. 4.9. Profiles of seats for teflon seals (a) and dimensions of the groove (b).

At a temperature of 50°C the failure of the rubber ordinarily takes place along the periphery of the groove. An increase in the width of the groove at a given linear load in this case promotes easier operation of the seal, i.e., increases the service life. Thus, an increase in the width of the groove from three to six mm increase the service life for closed seats of type I (see Fig. 4.8) by 3-5 times (at a linear load of ρ = 25 kgf/mm).

At low temperatures (around -40°C), when the rubber becomes more rigid and brittle, failure begins usually directly beneath the seat. In this case an increase in the width of the groove does not affect the moment of the start of failure. The permissible number of operations at low temperatures is minimal, i.e., these conditions are the most rigorous. But during operation the required number of triggerings at low temperature is small. A functional test of the assemblies (when, strictly speaking, the service life is being depleted) in the majority of cases occurs at plus temperatures.

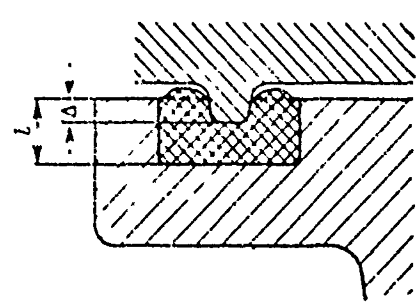

Fig. 4.10. Diagram of the compression of a seal by the seat.

However, an increase in the width of the channel leads to an increase in the dimensions of the seat of the closed type, which has an adverse effect on the overall size of the assembly or on its hydraulic resistance. Moreover, it should be kept in mind that at a high velocity of the gas flowing around a wide seal it can be torn from its seat in the case of insufficient strength of attachment to the metal. These considerations limit the width of the channel.

For a teflon seal the width of the groove should be at least 1 mm larger than the width of the seat (not less than 0.5 mm on each side).

The magnitude of compression affects the working capacity of rubber and teflon. By compression w we mean the ratio $(\Delta/l \cdot 100)\%$ where l is the depth of the seal, and Δ is the insertion depth of the seat (Fig. 4.10).

With prolonged storage of the seal with low compression the hermeticity may be lost.

In normally closed valves, when the seat is constantly imbedded in the rubber, long-acting high pressure leads to the destruction of the rubber. Under short-term loads (in normally open valves) the percentage of compression may be significantly greater. Thus, for a rubber seal the magnitude of compression during prolonged storage should be within the limits of 10-30%, and during short term action compression up to 50-55% is permitted. These data should serve as a guide for the selection of the depth of the channel.

For seals made from teflon the usual value of compression in normally closed valves amounts to 12-16%. For normally open valves greater compression is also not recommended, since in proportion to the operating service life the hermeticity of the joint will deteriorate in view of the residual deformation of the teflon.

4.1.3. Type and Profile of the Seat

Closed and open seats (see Fig. 4.8) are used in valves of LPhEs. A closed seat (see Fig. 4.8, type I) limits the deformation of the material of the seals on both sides: using it, long

service life is achieved, but the hermeticity is worse than when using an open seat of type II.

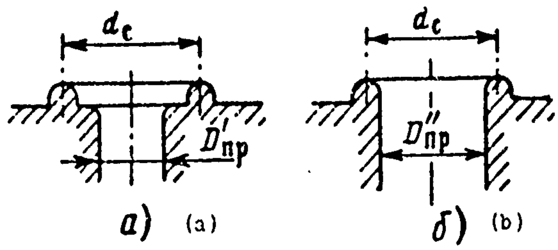

Fig. 4.11. Effect of the type of seat on the flow passage cross-sectional areas in the valve: a - closed seat; b - open seat.

A seat of closed type, in comparison with a seat of the open type of equivalent diameter d_c, has fewer flow passage cross-sectional areas: $D'_{\text{пр}} < D'_{\text{пр}}$ (Fig. 4.11). Therefore the use of closed seats results in either an increase in hydraulic resistance, or in the growth of the dimensions of the assembly.

If in closed seats the inner diameter of the seat $D_{\text{пр}}$ exceeds the inner diameter $D_{\text{вн}}$ of the rubber seal (Fig. 4.12a), then due to the presence of the sharp edges A in this case destruction of the rubber is possible. It is therefore essential that the inner diameter of the seat $D_{\text{пр}}$ be at least 0.2 mm, and preferably 1.0-1.5 mm less than the inner diameter of the rubber seal $D_{\text{вн}}$.

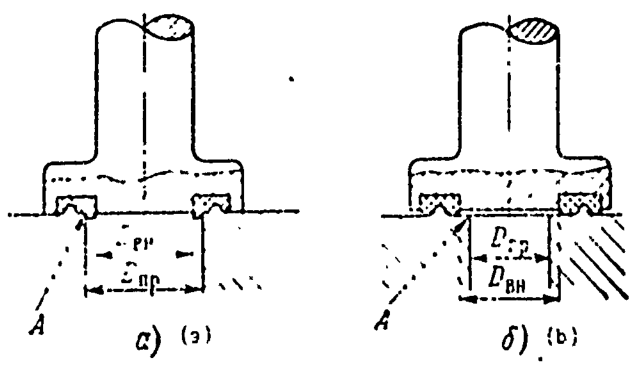

Fig. 4.12. Relationship of the dimensions in seals with closed-type seats: a - incorrect ($D_{\text{пр}} > D_{\text{вн}}$); b - correct ($D_{\text{пр}} < D_{\text{вн}}$).

In a closed-type seat, shown in Fig. 4.12b, strictly speaking, touching of the metal of the valve against the metal of the seat is impossible, since there will always be a layer of packing material between the valve and the seat. The higher the seat, the greater the stress in the packing material may be created, until the metal of the valve rests in its seat. The wider the seat (with identical height), the greater the volume of packing material is deformed. This leads to great stress in the rubber and, consequently, to large residual deformation. In seals made from teflon high seats should be avoided; the seats are made of the closed type and wide - so as to reduce the specific pressure, and low - so as not to allow considerable deformation. High sharp seats of the open type are in general employed only with very low linear loads.

Table 4.5

ρ_{max} kgf/mm	10	15	20	25
Number of operations	600	500	450	400

Remarks. The value ρ_{min} for this seat with a sealing pressure of 230 at must not be less than 0.8 kgf/mm.

In practice, an efficient seat shape, which ensures both hermeticity of the locking mechanism and the required operating service life, is selected either as a result of analysis of the operation of similar designs, which have approximately identical working conditions, or else by means of comparative tests of several variants.

For seats, the data of which are shown in the second column of Table 4.2, the maximum permissible number of operations depending on the linear load is given in Table 4.5.

4.1.4. The Pressure Magnitude and the Sealing Medium Properties

Locking mechanisms which have identical seat profiles, equivalent specific loads, identical sealing materials and so forth, will possess equal degrees of hermeticity depending on the value of the working pressure and the properties of the sealed liquid or gas.

Experimental data, obtained during tests in compressed air to determine the effect of pressure on the necessary linear load, are shown in Fig. 4.7. The unbroken line plots the assumed dependence $\rho = f(p)$, and the points give the actual data, obtained with various seat heights.

With the change in the pressure, as a rule, the linear load on the packing material itself also changes (Fig. 4.13): with a decrease in pressure (see Fig. 4.13c) it increases, and in the diagram shown in Fig. 4.13a - it decreases. In the cases shown in Fig. 4.13b, c and d, the hermeticity is improved, while in the case shown in Fig. 4.13a - it worsens.

The seal hermeticity depends in large degree on the properties of the sealing gas or liquid. The greater the viscosity of the sealing gas (liquid), the greater, as a rule, is the degree of hermeticity that can be achieved. However, even though the viscosity of kerosene is greater than in water, hermeticity is much more difficult to ensure in kerosene.

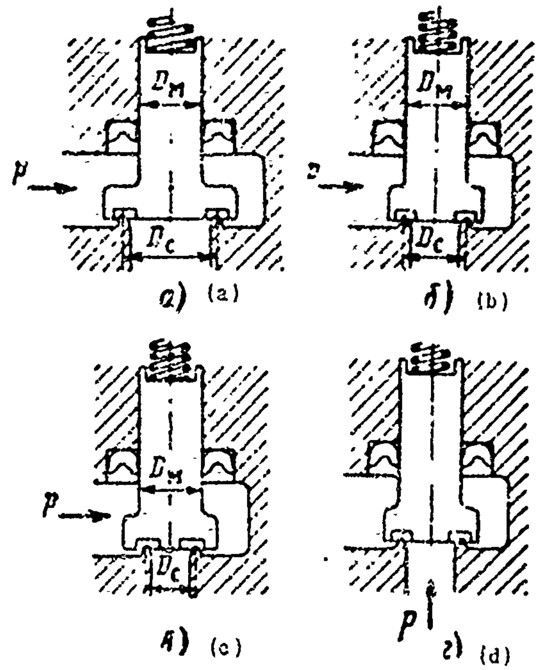

Fig. 4.13. Effect of the change in pressure on the sealing load: a - $D_M < D_c$ (with an increase in pressure the load increases); b - $D_M = D_c$ (with an increase in pressure theoretically the load does not vary); c - $D_M > D_c$ (with an increase in pressure the load is reduced); d - with an increase in pressure the load is decreased.

Helium is the most fluid gas; it is somewhat more difficult to ensure a high degree of hermeticity of the locking mechanism when handling helium than when using another gas. (Helium is capable of penetrating through intact rubber in insignificant quantities).

In practice it is very important to establish the dependence between hermeticity of the unit under atmospheric pressure and when operating with the employed propellant. With such a dependence a check of the assembly with compressed air could give an evaluation of the degree of hermeticity of the unit when handling propellant. However, unfortunately, a strict law,

applicable for various diameters and seat profiles, has not been established to this time. Depending on the individual peculiarities of the unit, on the workmanship of the seat profile, the machining finish and so forth, this relationship is quite different. Therefore, when total hermeticity of the locking mechanism handling propellant is required, attempts are also made to ensure total hermeticity under the same conditions when handling air or, in the extreme case, a very small air leakage is permitted.

4.1.5. The Surface Finish on the Seal and of the Seat Profile

During the machining operation the rubber should not be overheated, there should be no mechanical strains or surface damages on it, because this has a significant influence on the hermeticity packing. The rubber is machined with a undant cooling from the center of the packing on both sides "into the metal," so that under the effect of the cutting force the rubber is not separated from the channel, i.e., so that the cement works for compression, and not for expansion (Fig. 4.14).

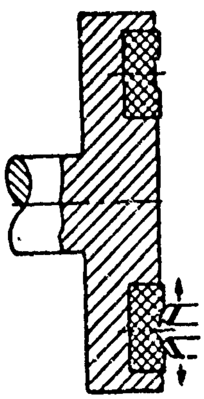

Fig. 4.14. Diagram of preliminary working (machining) of a rubber seal.

Final lathe processing is conducted from the center of the part to the periphery (by special cutters) at high velocities and at low feed rates; the projection of the top of the rubber beneath the surface of the metal reaches as high as 0.05 mm (due to the

compression of the rubber during machining). Working by grinding or by fine sandpaper is used.

The machining of teflon on lathes presents no difficulties. The degree of the surface finish of the seat has a significant effect on the operation of the seal. Defects in the seat lead, in proportion to the increase in the number of operations, to the appearance of cracks, breaks, "fraying" of the rubber and, as a result, to the loss in hermeticity of the unit. Radial indentations in the seats have an especially adverse effect on hermeticity.

In order to provide the seat profile with the required accuracy and finish (not less than ∇6, and in a number of cases ∇8), the seats are manufactured using a special shaped cutter, which machines the entire profile of the seat at once. The manufacturing precision of the cutter determines the precision of the profile of the seat.

The purity of the air or propellant passing through the assembly has an effect on the working capacity of the seal. In air or in the propellant there may be scale from the supply lines, or fine shavings, formed when the threaded connections were screwed on and carried away by the propellant. These extraneous particles have a destructive effect on the packing.

4.1.6. Character of Operations

The character of the operations is determined by the speed of movement of the valve to its seat, which depends on the value of the acting forces. The less frequent the triggering, the more hermetic will the seal be, but the more quickly can the destruction of the rubber or the wearing out of the teflon occur.

CHAPTER 5

THE SEALING OF NONMOVING JOINTS

The sealing of parts, which are stationary relative to one another, is accomplished in assemblies of LPREs in various ways. In pneumatic-automatic assemblies to seal stationary connections most frequently employed are flat metal gaskets made from a plastic material - copper, aluminum, more rarely rubber and still more rarely from teflon. In pyroautomatic assemblies the hermeticity of stationary joints is frequently provided by welding.

The hermeticity of joints with flat gaskets is achieved by the deformation of the material of the gaskets, flowing into the microunevennesses of the surfaces. Therefore, to ensure hermeticity, it is essential to create a certain pressing effort of the gasket to the sealed parts. The creation of the pressing force is accomplished in various ways: with the aid of flanges made integral with the basic parts (see, for example, the assemblies shown in Fig. 2.8 and 2.12), or with the aid of coupling flanges (Fig. 5.1); with flow passage cross-sectional diameters in assemblies of up to approximately 30 mm retightening of the gasket is usually accomplished using threads. In this case the thread is made either on both sealed parts[1] (Fig. 5.2a), or, (what is

[1]This method is worse, since turning of the gasket can occur, which impedes the assurance of hermeticity.

preferable) the compression of the gasket is ensured by coupling nut 2 (see Fig. 5.2b), which eliminates turning of the gasket.

Maximum specific pressures (in kgf/mm^2) for flat gaskets with a thickness of 0.8-3.0 mm, according to the data of G. V. Makarov [9], are recommended as follows: aluminum - 1020-1400; copper - 2520-3150; asbestos - 112-455; teflon - 112-434; rubber 28 (the smaller numbers refer to the thicker gaskets).

To ensure hermeticity, the specific pressure of the preliminary compression of the gasket to the basic parts (the contact pressure) should be 1.5-3 times higher than the pressure of the sealed medium. Where it is necessary to seal a gas, the contact pressure should be higher than to seal a liquid with the same pressure[1]. The hermeticity of the connection will be assured, if on the surface of the part the points at which the required contact pressure is achieved form a closed curve, or what is even better, a circular area (5.3a).

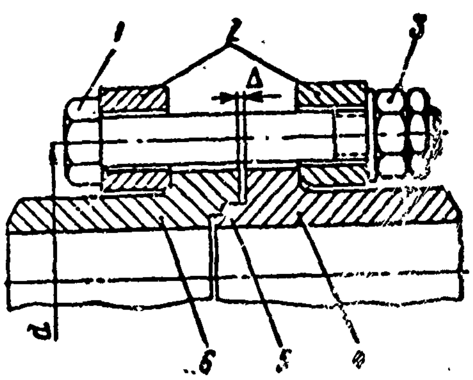

Fig. 5.1. The sealing of a gasket using coupling flanges: 1 - bolt; 2 - coupling flange; 3 - nut; 4 - nipple, 5 - gasket; 6 - connector fitting.

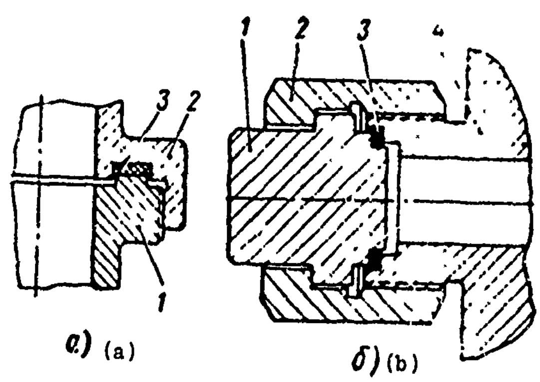

Fig. 5.2. Seal of the gasket using a threaded connection: a - thread on both sealed parts; b - thread only on one seal part; 1 - nipple; 2 - nut; 3 - gasket; 4 - connector fitting.

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[1]The contact pressure to create hermeticity with a seal of certain liquids, for example, kerosene, must not be less than that required to seal air with the same pressure.

To promote the creation of hermeticity for the gasket, if possible under operating conditions[1], a lubricant grease is applied. The lubricant fills the minute unevennesses and depressions, which exists in the surface of the parts. Even a very insignificant lubricant layer very substantially increases the hermeticity of the connection. Therefore, the creation of lubricant greases, suitable for application even if only in the stationary joints (i.e., working under relatively light conditions in comparison with the rubbing pairs), and stable with corrosive oxidizers, is very essential.

To provide hermeticity of the seal by means of a gasket during prolonged storage of the assembly, the gasket should be put into special channels (so-called placement of the gasket "into a lock", Fig. 5.4a and b). With such a design the gasket remains in a closed volume. This excludes the possibility of "flow out" of the gasket with the passage of time, which can occur with a gasket that is simply pressed between two parts (see Fig. 5.4d). The flow out of the gasket leads to a weakening of contact pressure (by the removal of stresses in the parts) and to the loss in hermeticity. This can also occur in a seal of the type shown in Fig. 5.4c. A seal of the type shown in Fig. 5.4a is more reliable.

The size of the channel under the gasket when made as shown in Fig. 5.4b is selected with the intent of the complete filling of the channel by the gasket material. In this case the cross-sectional area of the gasket exceeds the area of the channel cross section by 20 to 30%.

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[1]Sometimes it is impossible to do this because of the danger of explosion or the inflamation of the lubricant when it comes in contact with the propellant, for example, with oxygen.

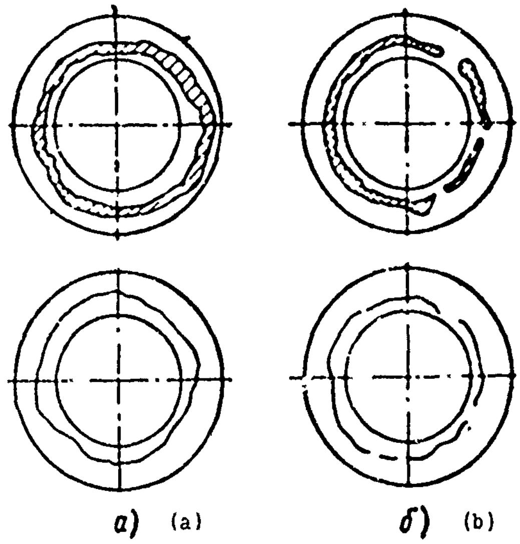

Fig. 5.3. Distribution of contact pressure over the sealing surface: a - correct (points with required contact pressure form a closed curve or a circular area); b - incorrect [the curve (area) of the points with the required contact pressure is broken].

With thin gaskets (0.3-0.5 mm in thickness) it is necessary to see to it that the parts have annular groove marks, into which the gasket material will flow. It is desirable that the grooves do not coincide on the mated parts (see Fig. 5.4e). Sometimes a "tooth" is made against the grooves (see Fig. 5.4f). In all cases the axial dimensions of the parts should ensure the presence of clearance Δ (see Fig. 5.4a) between the surfaces of the cover and of the housing. Clearance Δ should not exceed 0.5-0.6 mm.

With large diameters of the flange connection the constancy of clearance Δ should be checked with a feeler guage over the entire perimeter of the connection.

It is desirable that the axes of the bolts 1 (see Fig. 5.4) be as close as possible to the gasket - this reduces the deformation of the flanges.

Lateral clearance when installing the gasket into the channel of any design should be minimal; with a seal design of the type shown in Fig. 5.4a, the diametric clearances between the housing and cover should not exceed 0.1-0.3 mm (on a side).

The recommendations given refer to copper and aluminum gaskets; however, also when using flat gaskets made of rubber or teflon in the channels the considerations stated above remain in force. To facilitate the achievement of hermeticity metal gaskets (chiefly copper) are frequently cadmium plated. The layer of cadmium, a soft metal, plays the role of a unique lubricant. Moreover, and this is the main thing, the layer of cadmium prevents the formation of a voltaic couple, leading to the corrosive destruction of the part, between the copper and the metal of the part.

A snortcoming of the above described flat metal gaskets is the fact that in this case it is necessary to ensure the plastic deformation of the material of the gasket; this requires significant force, which leads to an increase in the dimensions and weight of the joint. In view of the low elasticity of the gasket in the compressed state, it is difficult to compensate for its possible misalignments (especially with large diameters). The connection must possess adequate stability, so as to eliminate the bending of the flanges.

Basically new types of seals have recently been developed. These have been widely used mainly in the USA and in France. Let us point out a few of them.

At low pressures, on the order of 35 at, corrugated gaskets are employed (Fig. 5.5). The thickness of the gasket material is 0.25-0.80 mm (see work [9]). The use of corrugated gaskets permits a sharp decrease in the force required to ensure reliable sealing, and a reduction in the loads acting on the joints. According to literature data (see work [21]) the tightening force of corrugated gaskets, in comparison with the tightening force of flat gaskets, is reduced by 5-10 times.

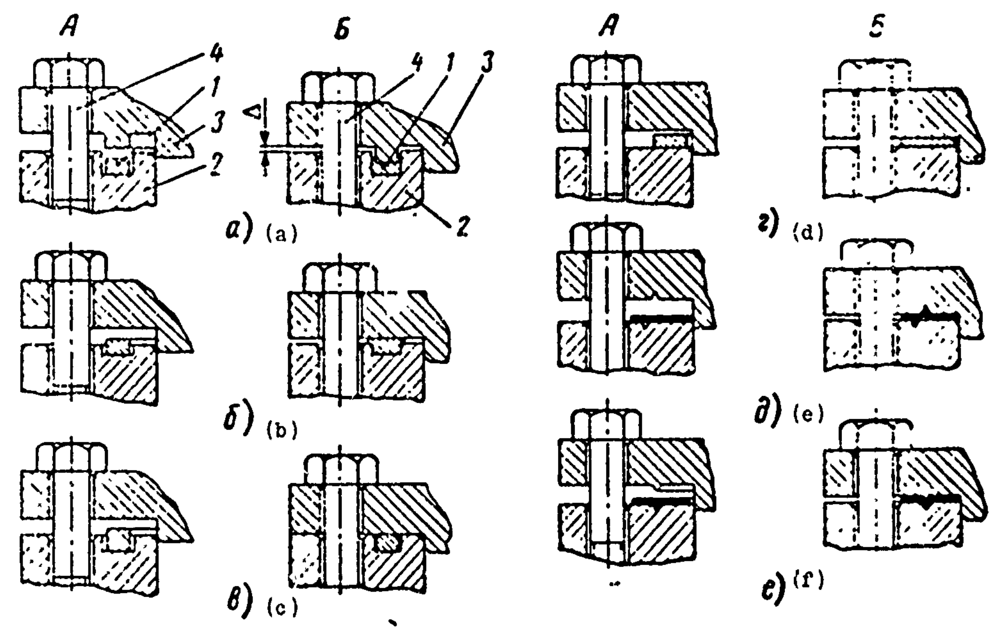

Fig. 5.4. Various methods of installing gaskets: a - position before tightening the gasket; Б - position after tightening the gasket; 1 - gasket; 2 - housing; 3 - cover; 4 - bolt.

The effectiveness of the seals depends on the gasket material, the operating temperature, the type of working medium, the machining finish of the flanges (a high degree of finish is not obligatory her.). Also employed are two- and three-layer corrugated gaskets. This permits an increase in the pressure of the sealed medium up to 70 at. For higher pressures spiral-wound gaskets (Fig. 5.6), shaped metallic gaskets (see Fig. 5.8 below) and gaskets made of "composite" materials are employed, using their high elasticity with relatively low tightening forces.

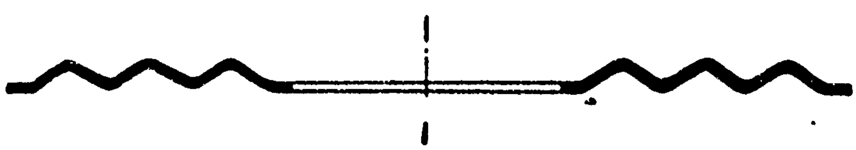

Fig. 5.5. A corrugated gasket.

Spiral-wound gaskets are manufactured by winding 14-15 layers of thin (0.17-0.2 mm in thickness) contoured V-shaped or M-shaped band, usually made from steel but sometimes from phosphorous bronze, copper and so forth. Between the layers of the band there is a filler - most frequently teflon or asbestos. The filler layer is 0.25-0.8 mm thick. In tightening a connection the profile of the band creates elasticity. The deformation during tightening may reach as high as 45% of the gasket thickness. The elasticity of the gasket is regulated by the density of the winding. The usual gasket thickness (width of the band) is 3.18-4.76 mm. The highest elasticity is possessed by M-shaped gaskets, employed in cases where significant stress of precompression is not permissible.

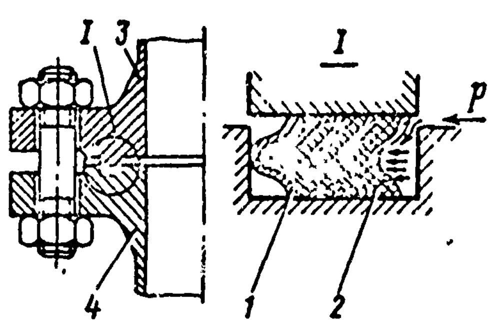

Fig. 5.6. Diagram of operation of a V-shaped spiral-wound gasket: 1 - band; 2 - filler; 3, 4 - flanges.

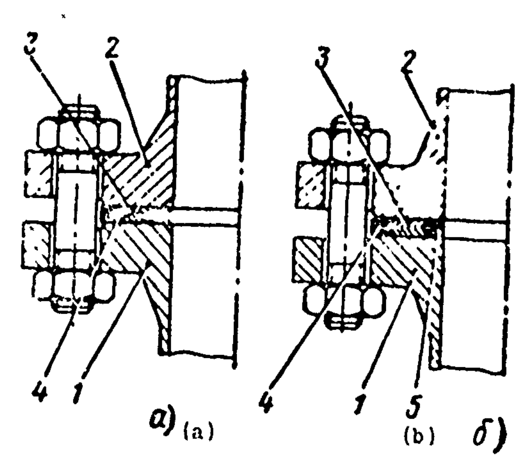

Fig. 5.7. Sealing assemblies using spiral-wound gaskets with flat flanges: a - with outer spacer ring; b - with inner and outer spacer rings; 1, 2 - flanges; 3 - gasket; 4 - outer spacer ring; 5 - inner spacer ring.

Spiral-wound gaskets can be conveniently installed in channels of flanges (see Fig. 5.6) (the installation "into the lock").

With low pressures flat flanges are also used with special spacer rings - either with outer (Fig. 5.7a), or with inner and outer spacers (see Fig. 5.7b). The working pressure still further increases the compression of the gasket against the flanges, while the filler fills the micro-roughnesses of the surfaces. These gaskets are used with pressures up to 175 at, with pulsating loading, with abrupt pressure change, during vibrations and so forth.

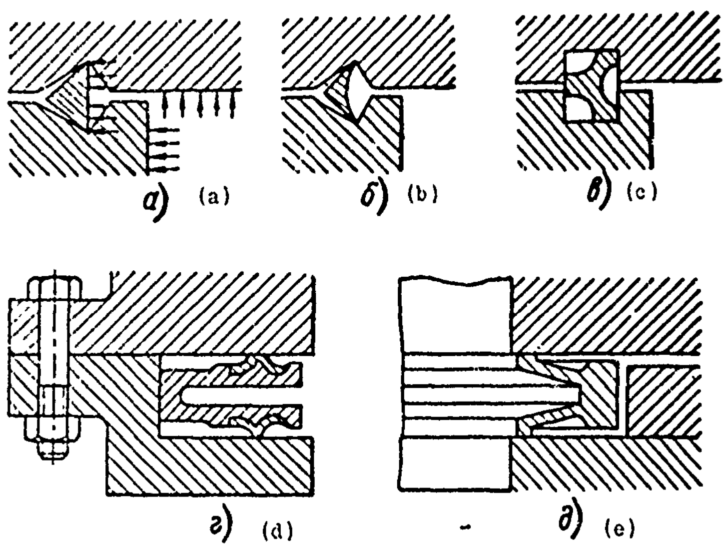

Fig. 5.8. Connections: a, b, c, - with shaped metal gaskets; d, e - with spring gaskets.

A so-called gasket made from "composite" materials, or from a "metallic rubber", is an intertwined, compacted steel wire with a diameter of 0.025-0.175 mm (a segment with a link of 3-19 mm), coated with teflon or silver, tin, rubber, copper, etc. The springy properties are ensured by the wire, and sealing characteristics are ensured by the filler material. Billets of such "metallic rubber" are machined to given gasket sizes. Sometimes a wire is manufactured from a billet; this wire is then butt-welded, producing a certain gasket shape. Such gaskets operate at pressures of up to 350 at, at high (500°C) and low (-195°C) temperatures, undergoing many stress alternations.

Connection designs using special shaped metal gaskets are shown in Fig . 5.8a, b and c. Such connections are designed to operate at high pressures - on the order of 1000-2000 at. The hermeticity of the joint when using such a gasket is increased with the increase in pressure - these are self-sealing designs.

The spring rings depicted in Fig. 5.8, d and e, represent a further development of the above-described gaskets, intended for high pressures. The pressure, entering inside the ring, expands the elastic plates of the ring. The outer diameter of the rings, shown in Fig. 5.8d, must be centered in the cannelure of the housing, but for the joint shown in Fig. 5.8e, due to the massiveness of the ring periphery, this is not essential, and this simplifies the manufacture and assembly of the unit.

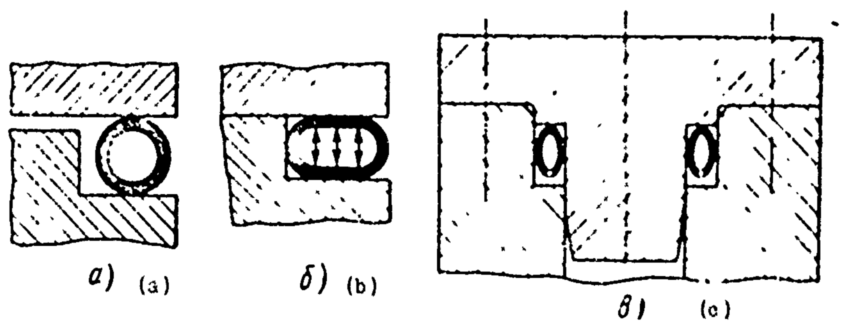

Fig. 5.9. Sealing by hollow metal rings: a - before tightening the joint; b - after tightening the joint; c - design example.

According to literature data, such seals are found in use (depending on the material) at temperatures of -252°C-+1200°C and at pressures of up to 3000 at (see work [20]).

Materials for such gaskets are chrome-nickel steels, refractory alloys, for example, inconel, ordinarily with a covering - of copper, silver, gold, nickel or a fluoroplastic.

Widespread use is enjoyed by hollow metal rings, which operate at high pressures and temperatures (Fig. 5.9).

There are the following types of rings: I - rings made of tubes, which are butt-welded; II - gas-filled rings; III - self-locking rings.

The operating scheme of the rings is shown in Fig. 5.10. The ring is thrust into the channel, creating reaction forces (contact pressure).

The most widespread are rings of the I type, employed at pressures p up to 700 at and temperatures from -250 to +1650°C. A thin-walled tube (minimum wall thickness of 0.13 mm) is given the shape of a channel (usually a circular form, however this is not necessarily so) and then the ends of the tube are butt-welded. Ordinarily the outer diameter of the tube employed is selected from 3 mm and higher (there are seals with a tube diameter of 12.5 mm), while the diameter of the joint ranges from 100 mm and higher. When assembling the connection the tubes are subjected to preliminary compression within the limits of elastic deformation; the diameter of the tube is compressed 20-30% in height. Such connections are effective for extended service and storage lives. According to American data a check of the connection, after it had remained in sea water for nine years, produced good results.

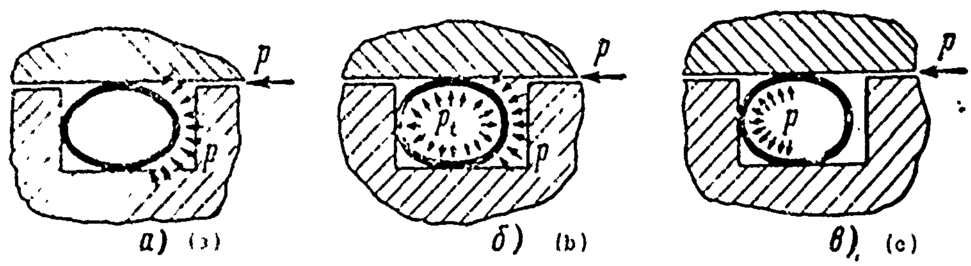

Fig. 5.10. Diagram of operation of hollow metal rings: a - hollow ring (type I); b - gas-filled ring (type II); c - self-locking ring (type III).

The required tightening force when using similar rings is reduced approximately by 10 times in comparison with the tightening force of flat gaskets.

Gas-filled tubes (type II) are welded in an atmosphere of compressed gas and are employed at high temperatures and relatively low pressures. In heating (during operation) the gas in the tube expands, its pressure increases to a value p_t and because of this increases the degree of compression of the tube against the channel walls.

Self-locking rings (type III) are ordinary rings of type I, in which there are openings from the side of the compressed pressure (see Fig. 5.10c). Then pressure p is supplied inside the tubes; this pressure expands the tube some more, improving the seal. Such rings operate at pressures of 3500 at and higher. However, as a result of the difficulty of cleaning the inner surface of the tube the use of such rings for corrosive propellants is limited. In these cases spring gaskets of the type shown in Fig. 5.8d are used.

Of the nonmetallic gaskets, rubber rings and gaskets made of teflon are used in the automatic equipment of LPREs.

For stationary joints (Fig. 5.11) the percentage of compression of the rubber rings must be within the limits of 25-50%.

In certain cases in automatic equipment of LPREs a seal is created during the installation of the tubing without using gaskets - by means of compressing the material of the parts themselves.

A nipple connection, shown in Fig. 5.12, is used with a propellant or gas pressure of up to 200 at. The diameter of the tubing cross section may here reach 30 mm, and at lower pressure

even higher than 30 mm. Sealing is achieved by tightening connector nut 2 (see Fig. 5.12).

With pressures exceeding 200 at in the assemblage of the tubing seals with gaskets are employed (Fig. 5.13). With even greater pressures - up to 1000 at - and tubing diameters exceeding 30 mm lens connections are used (Fig. 5.14).

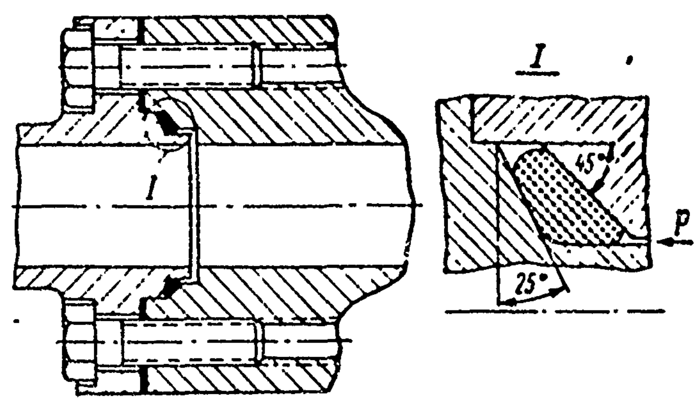

Fig. 5.11. Seal using a round rubber ring in a wedge-shaped opening.

With any type of seal the necessary degree of tightening of the connection must be established. In practice, it is very difficult to select such a tightness when assembling the connection in the workshop, which would guarantee the hermeticity not only at the given moment, but even after storage of the assembly, and under the forthcoming operating conditions. Ordinarily the tightening value is established as a result of experimental adjustment.

The degree of tightening is measured in various ways; from the applied torque (using a calibrated torque wrench); from the elongation of the bolts during tightening; from the angle of rotation of the nut during turning, measuring from the moment

of contact of the parts. All of these methods have various disadvantages, which limit their use.

The achievement of a given torque during tightening still does not guarantee obtaining the specified tightening force, since the torque value is affected by the quantity of lubricant carried on the threads of the connection and on the gasket, as well as by the purity of manufacture of the threads.

Measurement of the tightening (elongation) of the bolt is theoretically a precise method, however in practice it is very difficult to measure the elongation of bolts.

Most frequently the degree of tightening is specified by designating a certain range of the angle of rotation of the nut (tightening by 1-1.5 facets, tightening by 2.5-3 facets and so forth). The difficulty with this method is in determining the moment of the beginning of counting, i.e., that moment when the mated surfaces come in contact with one another and the entire axial clearance between them is eliminated.

One of the new methods of ensuring hermeticity of joints, which operate at low excess pressures (not greater than 2-2.5 at), involves the use of various types of cements and sealing compounds, using which the parts are "glued." The best results are produced by: epoxy lute, sealing compound UT-32, compound K-115, compounds K-168, K-153 and others.

Epoxy lute is a dry powder, consisting of epoxy resin, filler, a plasticizer and a dry pigment (coloring agent). This powder is dissolved in a liquid solidifier, which is a 50% solution of hexamethylenediamine in alcohol. The solution is made not more than 1.5 hours before the cementing process.

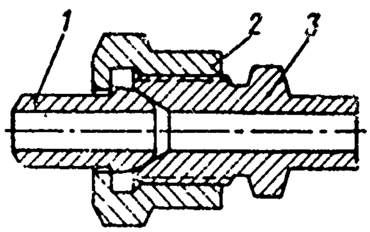

Fig. 5.12. Nipple connection:
1 - nipple; 2 - connector nut;
3 - connection tube.

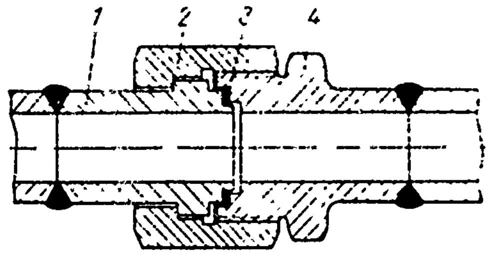

Fig. 5.13. The connection of tubing with the use of a gasket:
1 - nipple; 2 - connector nut;
3 - gasket; 4 - connection tube.

The lute is applied using a spatula to the precleaned and degreased surface of the parts in a layer of 1-2 mm (to each part). The parts are joined together and are subjected to drying in the compressed state (sometimes using special equipment). After drying an elastic film remains on the surface of the parts. The luted joint is operable within a temperature range of $\pm 50°C$. The lute may be applied for the joining of not only homogeneous materials but also for cementing aluminum to steel, rubber to metal and so forth.

The storage time - without loss in operability and strength of the connection - is eight years.

Compounds are made up on a basis of epoxy resins ED-6 or ED-5. Thus, the K-115 compound is the ED-5 resin, plasticized by polyester MGF-9. Polyethylene polyamine serves as the solidifier.

The cementing strength possessed by the compounds depends on the material of the cemented parts. For example, the cementing strength of aluminum alloy D_1 with fiberglass laminate at 15-20°C is 150 kgf/cm^2 (shear force).

The usability period for the compounds after their manufacture (after solution in the hardener) is not great - it is approximately 1 hour. After drying the connection becomes strong and can be stored for years, without losing its strength. A connection using a compound is stable in a medium of oxidizers based on nitric acid as well as in a medium of various forms of fuels.

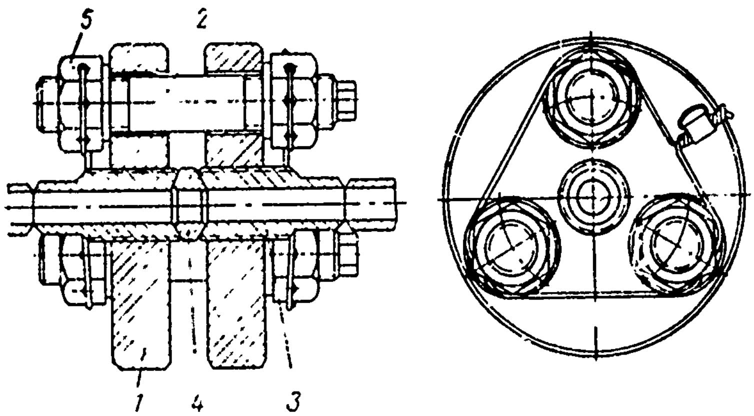

Fig. 5.14. Lens connection for tubing: 1 - flange; 2 - bolt; 3 - threaded end (connector tube); 4 - lens; 5 - nut.

Cements and sealing compounds are also used for sealing threaded connections - strictly speaking, this is why they were created.

In the assemblies of LPREs (mainly in automatic pyrotechnic assemblies) welding has been used to connect parts or to connect assemblies with the tubing network. This method of connection ensures reliable design simplicity.

Because of this advantage, and also because of their capability of cooperating in an environment of corrosive components, welded connections are now being used more and more.

Argon-arc and electron-beam welding are employed to weld the parts. These types of welding avoid distortion and deformation of the parts, eliminate the formation of scale, and ensure fine, even and smooth seams.

Argon-arc welding is welding in an inert gas protective atmosphere - most frequently of argon, from which this method received its name. However, helium, and sometimes carbon dioxide, are also used as the inert gas. The welding is done by an electric arc between the welded material and an electrode.

Argon-arc welding is used in welding stainless and refractory steels of the austenitic class, aluminum alloys, copper, magnesium, and their alloys, and titanium and its alloys.

Argon-arc welding ensures high quality of the welded connections, especially for parts with a thickness of 0.1-0.5 mm, a reduction in the distortions and warpage of the material, and also permits mechanization of the process, which increases productivity. When welding alloyed chrome-nickel steels, widely used in assemblies of LPREs, this form of welding decreases the possibility of the emergence of intergranular corrosion, and because of this the operability of the assemblies is ensured after prolonged storage.

A more advanced method is welding by an electron beam using a so-called "gun." In electrode-beam welding a special device forms a narrow focused flow of electrons, directed onto the welded parts. The density of the energy of the electron beam is regulated within broad limits - up to 50-100 kW/cm^2 for the welding of thin parts up to 1000 kW/cm^2 for welding parts of great thickness. The device which creates the flow of electrons, just as the welded part, is placed into a vacuum chamber with a residual pressure of the order of 10^{-4} mm Hg.

The electron beam is easily directed and regulated. The zone of intense heating from the electron beam, according to foreign data, is a circle with a diameter of 0.2-1.0 mm (devices which produce a beam with a diameter of 0.08 mm and less are proclaimed abroad). Thus, only the very small zone of the welded part is subjected to heating. Heating and cooling of the part occur very quickly. As a result of this the possibility of distortion or warpage of the assembly is reduced to a minimum, and because of this high accuracy of the connection is insured and, moreover, the machinery for joining the parts before welding is significantly simplified.

Because of the fact that welding is done in a vacuum, the possibility of contamination of the welded seam is eliminated; the purity of the seam ensures high corrosional resistance of the welded unit.

Structural variations in the material of the near-weld zone with the described method of welding are minimal. Phenomena of intergranular corrosion are avoided when welding chrome-nickel steels as a result of the rapid temperature-change processes. A check of the rupture strength of the welded seam, produced by means of an electron beam, shows that the strength of the seam is very high[1], the fatigue properties of the seam are also high. The welded seam produced is not only strong, but also dense, which ensures a high degree of hermeticity of the connection.

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[1]The Zeiss firm recommends electron "guns," which produce a beam of micron thickness, oscillating at high frequency across the seam. Such a seam gives especially high strength indexes.

Electron-beam welding allows the possibility of joining materials with different thicknesses. The ratio of the depth of the seam to its width is 8-12 and higher. Moreover, electron-beam welding can be used to join heterogeneous materials. This method requires the mechanization of the welding process.

The disadvantages of the above described welding method include the relative complexity of the equipment, which requires the servicing of the machinery by highly skilled personnel. The presence of x-radiation during welding requires a plan for worker protection.

The widespread use of welded seams has brought about the necessity for changes in the design of assemblies (especially, one-time assemblies), in the technology of their manufacture and assemblage, and also in the procedure of processing and testing of the assemblies.

CHAPTER 6

CALCULATIONS AND TESTS OF PROPELLANT VALVES

6.1. BASIC CALCULATIONS IN DESIGNING PROPELLANT VALVES

In designing a new propellant valve the following tentative calculations are first carried out.

1. Calculation of the flow passage cross-sectional areas of the inlet and outlet tubes, determination of the path and the diameter of the locking mechanism.

2. Calculation of the value of hydraulic losses during steady-state conditions.

3. For pneumatic valves - calculation of the forces required to maintain the locking mechanism in the open position; calculation of the forces required to ensure hermeticity of the end seals.

For pyrotechnic-automatic assemblies - the determination of the type of explosive cartridges, based on the destructive force of the exploding element and the magnitude of the volume, in which the explosive cartridge is triggered.

4. Calculation of the valve with respect to strength - the determination of the required wall thickness of the housing and the dimensions of other basic parts.

5. The determination, based on the intended use and operating conditions, of the required operating service life of the pneumatic assembly.

Subsequently, after the refinement of the assembly design, check calculations are made. Having determined the masses of the moving parts and the acting forces, an equation for the motion of the locking mechanism is set up.

In calculating the flow passage cross sectional areas of the inlet and outlet tubes, or, in other words, in calculating the diameters of the propellant lines, we proceed from the permissible value for losses in static pressure Δp (in kgf/cm^2) in the communication lines at nominal propellant flow rate, determined by the speed of the liquid v (in m/sec).

The hydraulic resistance[1] of an element of the main line is expressed as:

$$\Delta p=\xi \frac{\gamma v^2}{2g}\cdot 10^{-4},$$

where γ is the density of the propellant in kg/m^3;

ξ is the coefficient of hydraulic resistance, depending on the local conditions;

g is the acceleration of gravity in m/sec^2.

[1]Strictly speaking, the hydraulic resistance and the pressure losses are not one and the same thing; the hydraulic resistance is losses in the energy of the fluid, which may be composed of losses of potential energy (the static pressure drop) and losses of kinetic energy (decrease in the dynamic head). If the velocities of the liquid at the inlet and at the outlet are identical (identical flow through cross-sectional areas), then the hydraulic resistance is equivalent to the difference in the static pressures at the inlet and at the outlet.

Values of the coefficient ξ for various cases of liquid flow (turning, drying, expansion and so forth) are given in handbooks on hydraulics.

The greater the value of Δp, the greater must be the pressure of the propellant at the outlet from the pumps with given flow rates and propellant pressures at the inlet to the combustion chamber and the greater, consequently, is the required power of the TPA (turbopump assembly). In order to avoid an excessive value of Δp and so as not to "oversize" the tubing, the propellant velocities in the main line are assigned an acceptable value and, by knowing the per-second flow rate G (in kg/sec) or Q (in m^3/sec), the area of the flow through cross section of the tubing F is calculated (in m^2) or, what is the same thing, the area of the branch connections of the propellant valve is calculated:

$$F = \frac{G}{\gamma v} = \frac{Q}{v}.$$

If the discharge of propellant from a valve is accomplished not through one, but through several tubes in parallel, then the value F is the sum of the flow passage cross sections of the branch connections of the valve.

The diameter of the seat of the locking mechanism of the valve is usually close to the diameter of the inlet branch connection tube. The lift (travel) h of the disk of the locking mechanism, to provide an equally great area of the flow passage cross sections, is taken as equal to not less than 1/4 of the inner diameter D_{BH} of the inlet connector tube. (Actually, in order to obtain the equality $\pi D_{BH}^2/4 = \pi D_{BH} h$ required that $h = 0.25\ D_{BH}$).

Calculation of the hydraulic resistances in the valve is carried out under steady state operating conditions - under main-stage and pre-stage conditions.

The velocity of the fluid inside the assembly frequently is assumed to be somewhat less than that in the lines, since there are in the assembly sharp turns of the flow, which lead to additional pressure losses. It is very important to create the correct configuration of the flow section of the valve, without permitting abrupt changes in the cross section and separation of the liquid flow, since this increases the coefficient of hydraulic losses.

If, before beginning calculation, there is already a sketch of the valve, then the pressure loss in the valve is calculated by determining the coefficients ξ_1, ξ_2, ξ_3 and so forth for each small segment of the liquid's path from handbooks.

The overall pressure losses in the valve are calculated as the sum of the losses in the individual segments:

$$\Delta p=\sum_{i=1}^{i=n}\xi_i\frac{\gamma v_i}{2g}=\frac{\gamma}{2g}\left[\xi_1 v_1^2+\xi_2 v_2^2+\ldots+\xi_n v_n^2\right].$$

Velocity v_i at a given flow rate is determined proceeding from the value of the flow passage cross-sectional areas in a given segment. In order that the formula be strict, it is essential to introduce a correction into a value of Δp for the difference in the velocity pressure heads: $\gamma(v_i^2-v_{i-1}^2)/(2g)$. However, ordinarily the error because of this is sc small, that it is disregarded.

Calculations of the forces required to control the propellant valves (closing, opening, transfer from one position to another)

present no special difficulties. These calculations are reduced: to the determination of the area, on which the propellant pressure (as well as atmospheric pressure) acts in various positions of the locking mechanisms; to the determination of the area, on which the pressure of air or of explosive gases must act; to the determination of the necessary air pressure in the controlling cavity or of the necessary weight of the explosive charge; and to the determination of the dimensions, rigidity and force of the springs in the assemblies.

The magnitude of the volume of the controlling cavity affects the triggering time of a pneumatic valve.

The value of the controlling pressure is ordinarily predetermined, proceeding from the possibility of using the existing pressure reduction valves and electro-pneumatic valves. It is advantageous to select as great a value as possible for the controlling pressure, since this reduces the area, and consequently, also the weight of the assemblies.

It is precisely during these calculations that the diagram of the valve should be refined. Depending on the calculated values for the area of the controlling cavity and the dimensions of the springs a final diagram of the valve (relief or nonrelief) is selected; the specific load for the sealing of the locking mechanism is calculated, and the possibility of using an assembly of the normally closed or normally open type is determined.

For normally open pneumatic valves we determine the minimum pressure in the controlling cavity, ensuring hermeticity of the locking mechanism at boost pressure and under other specific conditions; values are calculated for the controlling pressure, at which the valve begins to open and at which it is completely open for operation on prestage (or main stage) mode.

For normally closed pneumatic valves we calculate the controlling air pressures at the moments of the beginning of opening and of full opening of the valve, with consideration of possible divergences in the propellant pressures; we determine the minimum required value of the propellant pressure to keep the valve in the full-open position during main stage operating conditions and so forth.

For return valves the possible hydraulic impact magnitude is estimated.

In the first stage of valve design these calculations are of a tentative nature. However, once having introduced into the design definite dimensions of the seal rings, the seat diameter, the controlling air pressure and so forth, we will obtain definite principles for the opening and closing of the valve.

For explosive valves, depending on the strength of their explosive element, we determine the pressure of the solid-reacting gases, necessary for the opening (or closing) of the assembly, and then, knowing the requisite pressure and volume of the solid-reacting gases, we calculate the weight of the explosive charge, i.e., the typical size of the explosive charge. In determining the typical size of the explosive cartridge it is essential to consider the possible variance in the pressures of the solid-reacting gases, caused by the technical conditions of operation of the explosive cartridges. The design of the explosive valve must ensure its operability both at minimum and at maximum pressure, developed by the explosive cartridge.

The structural design of parts of propellant valves is ordinarily worked out for steady-state working conditions of the engine, allowing only for static load from the propellant and controlling gas acting pressures. The design of the valve body is worked out just like the design of a shell loaded by an

internal pressure. As the initial calculation pressure the designer should take not the working pressure in the assembly, but the higher value of the pressure of the hydraulic tests of the parts.

Ordinarily during such calculations the dynamic load from possible vibration during engine operation and from other causes is not taken into consideration. The greatest loads usually arise during the transient modes of engine operation, as a result of which they submit to calculation with difficulty and they are evaluated by means of selecting the appropriate margin of strength: often the strength, and consequently also the weight characteristics of an assembly in the initial stage of design are determined on the basis of existing experimentation and analysis of existing designs.

Dynamic loads are precisely the reasons for the breakdowns of assemblies or of their subassemblies, sometimes encountered during experimental tests of new engines. Of great significance is the dynamic load from vibration accelerations, directed along the axis of the combustion chamber; a constantly acting, gradually increasing overload from the increasing motion of the flight vehicle may have some slight effect. However, the greatest influence on the strength of the valves is exerted by nonsteady-state modes, mainly, engine start and stopping modes. At this time, because of a number of reasons (both general and specific for each type of engine) percussive loading of the valve parts takes place, hydraulic impacts in the fuel feed communication lines may take place, and vibrations may arise. It is usually very difficult to take all this into account in view of the complexity of the transient processes and the number of factors influencing them.

Causes which lead to a brief increase in pressure are examined below (in paragraphs a, b and c).

a) Impacts During the Closing or Opening of Propellant Valves

Hydraulic shocks, as a rule, always take place during the triggering of the explosive valves. Therefore, the tentative value for the increase in pressure, arising during the triggering of an explosive valve, is ordinarily assigned during calculation.

The value for the pressure increase Δp_r (in kgf/cm^2) in a complete hydraulic impact is in the simplest case calculated from the formula of N. Ye. Zhukovskiy:

$$\Delta p_r = \varrho a v \cdot 10^{-4},$$

where ρ is the mass density of the liquid (according to the MKGSS system) in $(kgf \cdot sec^2)/m^4$;

v is the velocity of the liquid before the opening of the valve in m/sec;

a is the rate of propagation of the shockwave of the pressure, i.e., the rate of propagation of sound in the liquid in the given tubing, in m/sec.

The value of a is determined as follows:

$$a = \frac{1}{\sqrt{\varrho\left(\frac{1}{E_1} + \frac{D}{\delta E_2}\right)}},$$

where E_1 is the modulus of the elasticity of the liquid (volumetric) in kgf/m^2;

E_2 is the modulus of the elasticity of the material of the tubing wall in kgf/m^2;

D and δ are the inner diameter and the thickness of the tubing in m, respectively.

As is known, the speed of sound a_0 in liquid (not limited by walls) is determined thus:

$$a_0 = \sqrt{\frac{E_1}{\varrho}}.$$

From a comparison of the formulas for a_0 and a it is clear that the velocity of propagation of a pressure wave in the tubing with a liquid is numerically close to the velocity of propagation of sound in this liquid. Both formulas coincide if in the formula a we assume that the tubing possesses maximum rigidity, i.e., $\delta \to \infty$.

Total hydraulic shock will take place in the case where the time of closing of the valve τ (the time of total deceleration of the flow) is less than the time of travel of the primary and reflected pressure waves in the tubing, i.e.,

$$\tau \leqslant \frac{2L}{a},$$

where L is the length of the tubing (in m) from the valve to the site of reflection of the shockwave (i.e., up to the mirror of the liquid, by joining the gas phase).

If $\tau > 2L/a$, then hydraulic shock will be incomplete, and in this case the pressure increase will be less; it is calculated thus:

$$\Delta p_r = \varrho a (v - v_1),$$

where v_1 is the velocity of the liquid in the tube at the moment when the reflected pressure wave reaches the closed valve.

There is in practice only one means to reduce Δp_{r}, i.e., to make the hydraulic shock incomplete - reduce the time τ, i.e., the time of total closing of the valve.

It is rather difficult to calculate the pressure increase (even for total hydraulic shock), since the above shown formulas are introduced for a cylindrical tube, while in an actual case we have a complex system of tubing and assemblies. To calculate the actual rigidity of the system is also not easy; the less the rigidity of the system, the more frequently (as a result of deformations) hydraulic shock will occur and the lower will be the pressure increase.

The dynamic forces in the body and in the locking mechanism, arising during the closing of the valve, are also a consequence of the acceleration during the movement of the moving system, this acceleration depends on the rate of change of the controlling air pressure and of the propellant pressure.

b) The Gasification of a Low-Boiling Propellant

With inadequate heat insulation of one of the segments of the propellant main line vapor lock may occur in it before the start of operation[1]. After the starting of the engine, at the moment when a gas bubble passes through the injectors, the shock of the column of liquid situated above this occluded gas occurs. With the passage of the gas bubble through the injectors, the speed of the flow increases abruptly (because for the gas the resistance of the injectors is significantly less than for the liquid); when the gas bubble has passed through the injectors, the liquid is

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[1]This phenomenona is sometimes encountered in test bench facilities when there is poor control of the condition of the heat insulation.

again decelerated and its speed in the tubing decreases sharply. The pressure increases with such deceleration can be calculated from the formula of hydraulic shock.

The greater the pressure differential at the injectors, the greater the gas cavity, the stronger the acceleration and subsequent deceleration of the flow, and the higher the value of the hydraulic shock. This hydraulic shock transfers to the propellant valve.

c) The Instantaneous Pressure Increase in the Combustion Chamber

With the combustion chamber pressure fluctuation at the moment of peak pressure there occurs an instantaneous cessation of fuel flow through the injectors - "blocking of the injectors" - as a result of the absence of a pressure differential on them, which leads to hydraulic shock. The hydraulic shock will be the stronger, the greater the per-second fuel flow rate, i.e., the greater the speed of the liquid. The magnitude of the pressure increase in the valve depends in practice on the position of the valve relative to the injectors.

As a result of the dynamics of any process leading to an abrupt, intermittant pressure increase p, vibrations of the valve body are excited. The stresses arising in the oody material, in the case of elastic deformations are proportional to the bending of the walls.

During axisymmetric vibrations, as follows from the work of I. A. Birger [3], the deflection of the walls w in time t during the period of action of pressure p is expressed thus:

$$w(t) = 2w_{\text{cт}} \sin^2 \frac{\pi}{T} t,$$

where w_{CT} is the static deflection of the walls under the action of the same pressure p (static);

T is the period of natural vibrations of the body.

The value of $w(t)/w_{CT} = k_Д$ is called the dynamic strengthening factor. The maximum value of the dynamic strengthening factor $k_Д$ may differ - it may be either greater or less than 1; this depends on the rate of increase and the duration of application of pressure, and also on the frequency of the natural vibrations of the body, however in any case the value of $k_Д$ may not exceed 2 (see work [3]).

Sometimes with the aim of the unconditional assurance of the strength of the assembly the calculation is made proceeding from the value $k_Д = 2$, i.e., assuming that the peak deflection value (or, what is the same thing, - the peak stress value) exceeds by two times the deflection (stress) from static loading.

Precise calculations of the acting stresses with consideration of the actual processes are rather complex, and to determine the strength reserves (with respect to the static pressure) after manufacture of the first samples of the assemblies they are tested with hydraulic pressure until destruction.

The determination of the required service life of a valve is made by analyzing the operating conditions of the engine, the conditions of its storage, and also by analyzing methods of monitoring its working capacity (this refers, naturally, to repeat-action assemblies). It is necessary to total the number of operations necessary in checking the valve in the assembly shop of the manufacturing plant, in installing the engine plant in the flight vehicle and so forth. For multiple-start engines one must calculate the number of possible starts.

The assembly design must be evaluated from the viewpoint of the possibility of ensuring the required service life. Naturally, such parameters do not lend themselves to strict calculation. The value of the possible triggering service life of a new valve may be tentatively estimated by studying an experiment of the operation of an analogous assembly, similar in design.

In conclusion, one should note the following. Having made preliminary calculations and experimentally checked the manufactured samples of the valve in autonomous tests, it still can not be unconditionally assumed that the given valve can be used in the given engine model. This conclusion can be drawn where, and only where, the valve has been tested on the engine, and provided the normal passage of the transient conditions and all the requirements imposed on the engine. The specifics of the transient processes may be such, that they require a review of the kinematics of the valve and a change in its rate of opening or closing, and this, in turn, requires more or less significant modifications of the design.

6.2. TEST BENCHES AND EQUIPMENT FOR TESTING VALVES

In the examination of the design calculations of the valve it was noted that their final modification takes place during experimental checking of the valve's operation while it is being tested. In the calculations it is impossible to determine with sufficient accuracy the value of the pressure differential on the valve disk, and to take into consideration the effect of friction, the effect of temperature, of the configuration of the parts, method of manufacture and others. Therefore, in the process of design refinement propellant pneumatic and explosive valves are subjected to a whole series of tests, such as: check of the hermeticity of the seals in air and in the components; determination

of the value of the pressures in the controlling and in the propellant cavities at the start and at the termination of movement of the locking mechanism; determination of the possible service life in the environment of the propellant component; determination of the hydraulic characteristics of the assembly; check of the design strength; check of the working efficiency in the entire operating temperature range; check after the vibration tests to simulate transportation of the engine; vibration stability and vibration resistance checks and so forth. The final check of the assembly is made during engine testing.

Not one of the forms of tests can replace another - each type of test reveals new aspects of operation of the assembly. A check of the assembly during its operation on the engine, although it is conclusive in the cycle of studying the assembly, all the same cannot eliminate the remaining checks: in the tests of the engine we ordinarily succeed in measuring all the assembly parameters which interest us, in clarifying all the peculiarities of operation of the valve, or in studying it from all aspects. The value of the valve tests on the engine consists in the fact that here not autonomous, but "circuit" requirements for the assembly are being checked (see Chapter 1).

The series of tests of the assembly on the engine allows us to solve the chief problem - concerning the fundamental possibility of using a given type of valve on a given type of engine, without touching on the important, but more particular problems, such as the assurance of hermeticity, the service life and so forth.

All the above-enumerated calculations are checked and are confirmed by independent valve tests.

1. The magnitude of the hydraulic losses is determined during a water-flow test of the assembly on a special hydraulic test bench. During the flow test the conditions in large degree approximate the actual operating conditions of the assembly; in these tests the magnitude of hydraulic resistances of the individual elements and of the entire assembly is determined, the direction and distribution of the flow of liquid and its effect on misalignments and the jamming of parts are investigated and so forth. The taking of the hydraulic characteristics is of great significance for understanding the operation of the assembly under various conditions (a special section, 6.3, is dedicated to this question). During the hydraulic tests it is possible to determine the time of triggering of the valves, as well as the tentative value of the pressure peak. It is impossible to simulate precisely the processes of hydraulic shock during the tests on the hydraulic test bench.

The accuracy of determining parameters during the hydraulic tests is usually lower than the accuracy of their determination in pneumatic tests.

2. The experimental determination of the magnitude of actual stresses in the pneumatic assemblies is made both when compressed air is supplied to the liquid cavity of the valve (in the pneumatic tests), and during the water-flow tests of the valve.

Pneumatic tests of repeat-action assemblies allows us to determine the controlling pressure of the beginning of opening of the valve, as well as the pressure of total opening with various pressures in the propellant cavity. During these tests it is possible to achieve high measuring accuracy, and to determine the influence of temperature, the influence of wear, manufacturing precision and the degree of finish of the parts and so forth, but in the pneumatic tests it is impossible to take into consideration the effect of the propellant pressure differential between the inlet

and outlet of the assembly, and consequently, it is impossible to take into account the dynamics of the process of the valve's opening. Thus, pneumatic and hydraulic tests do not eliminate one another, but rather supplement one another.

3. The strength tests of automatic assemblies of LPREs are usually conducted under static conditions, since it is very difficult to simulate the conditions of dynamics[1]. In the hydraulic strength tests it is necessary to simulate the heaviest load (forces only from the pressure of the controlling air, or forces only from the pressure of the propellant, or forces with the combined effort of the controlling air and of the propellant) - for different assemblies either this or that case of loading may be most dangerous.

In order to determine the deformations or stresses in the parts during the hydraulic tests, strain gauges are set up at the most appropriate places. Similar strain gauges are sometimes fastened to the assemblies during their operation on the engine. A comparison of the readouts of the strain gauges under dynamic loading and in static tests makes it possible to judge the actual strength reserves of the assembly.

4. The value of the assembly's triggering service life is checked during the tests on the special bench, permitting the simulation (with respect to pressure, rate of opening and closing, temperature and so forth) of the valve's operating conditions. Simultaneously, during such tests, in proportion to wearing out of the service life, the change in hermeticity of the seals and the effect of change in the temperature conditions on the hermeticity

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[1]In certain instances, however, it is necessary to conduct tests with the simulation of the dynamic loading.

are checked. Only on the basis of such carefully conducted tests is it possible to judge the true service life of the valve.

The conducting of the above cited tests requires special equipment, permitting:

- the assurance of a constant (fixed) air flow rate (gas rate) or the maintenance of a constant given pressure, where there must be a possibility of designating this pressure within broad limits;
- the automatic assurance of the given temperature of the environment;
- both abrupt (impact) air supply to the tested assembly, as well as a slow, smooth air supply; the possibility of both abrupt and smooth gas discharge;
- the assurance of the possibility of determining the hermeticity of the sealing sites;
- the determination of the effect of alternating forces, causing a change in acceleration (vibration, jolting);
- the checking of the operation of the assembly with a given constant direction of the acceleration forces;
- the maintenance of the assembly for a prolonged period under pressure of the propellant and a check of its operation with the presence of propellant in the assembly;
- the conducting of water-flow testing of the assembly, with the provision of fixed flow rates, and in a number of cases also with given liquid pressures.

Tests using compressed air are carried out on special pneumatic stands. The fundamental diagram of the simplest pneumatic test bench is shown in Fig. 6.1. Compressed air (nitrogen) is fed to the controlling cavity of the tested pneumatic assembly through tubing of line I or II; during operation air is fed to line II either slowly and smoothly, opening valve I behind the reducer (with EPV 2 previously opened), or suddenly (abruptly), triggering electropneumatic valve 2 when valve 1 has been previously

completely opened; along line I high-pressure air may be supplied. Along line III pressure is applied to the propellant cavity of the valve. The presence of lines I and II, on one of which a reducer is placed, when there is an EPV, behind which discharge jets can be installed, enables the simulation of practically any principle of supply or discharge of the controlling pressure.

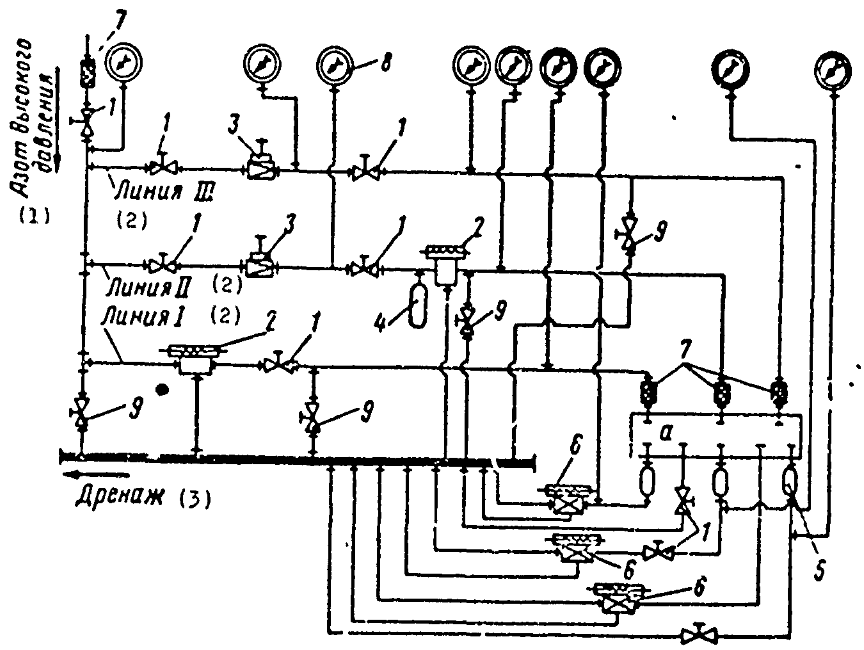

Fig. 6.1. Fundamental pneumatic circuit of the test bench: 1 - shut off valves; 2 - normally open EPV with drainage; 3 - pressure reducers; 4, 5 - buffer volumes; 6 - normally closed EPV with drainage; 7 - filters; 8 - manometer; 9 - drainage valve; a - location of the tested assembly.
KEY: (1) High pressure nitrogen; (2) Line; (3) Drainage.

Different types of discharge lines (in Fig. 6.1 there are five of them shown) may be connected to the outlet of the tested assembly.

Fig. 6.1 does not show the system for the assurance and maintenance of the required temperature conditions for checking the assembly. The required temperature of the environment surrounding the assembly is provided by various means.

A diagram of the simplest device for providing a given temperature, built into the test bench, is shown in Fig. 6.2. With the aid of fan 3 air circulates through the closed volume - the test chamber, formed by pan 2 and hinged cover 1. With the compression of the cover by the fast-acting coupler to the pin of the test stand the required adhesion tightness and sealing of the cover is ensured. To ensure easy cover opening there is a counterweight (this is not shown in Fig. 6.2). The cover and the pan must be armored and must have double walls, between which Mipor or felt, asbestos and so forth is inserted for heat insulation.

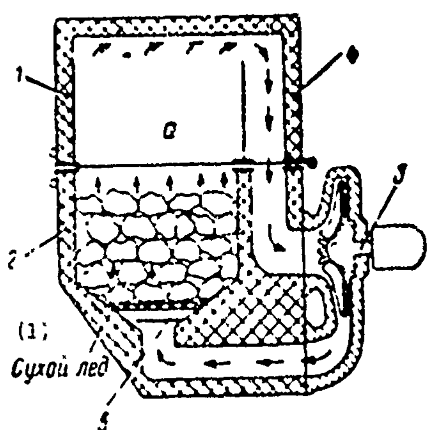

Fig. 6.2. Diagram of the device for ensuring a given temperature: 1 - cover; 2 - pan; 3 - fan; 4 - heat insulation; 5 - preheater; a - location of the tested assembly
KEY: (1) Dry ice.

In the upper part of the test chamber the tested assembly is installed with the aid of special brackets. Compressed air tubes, drainage devices and electric cables lead from the test bench to the assembly.

In the lower part of the pan, in the channel for the passage of air, preheater 5 with a power of about 1 kW and operating on

a voltage not exceeding 35 V is set up. There is a grid above the preheater.

To conduct tests at elevated temperatures (on the order of 50-60°C) the preheater is switched on. Warm air, rising upward, ensures the required temperature. To ensure uniformity of the temperature field the fan is turned on from time to time. When the given temperature is reached, the heating is switched off by means of a thermal regulator.

For low-temperature tests (not lower than -60°C) crushed dry ice is placed into the grid. When the fan is switched on, the air circulating in the closed circuit, passes through the dry ice and is cooled.

Checking of the valve "for jolting" is accomplished on special test benches, which create intermittent overloads on the order of 9-10 g, of the type shown in Fig. 6.3. The tested assembly is fastened to the table of the jostling stand. This table moves in a vertical direction with a certain frequency.

A diagram of one of the jostling stands is shown in Fig. 6.4. lever 1, sliding on cam 2, which is rotated by an electric motor, raises table 3, compressing springs 4 and 5. The cam profile from point A to point B ensures a uniform raising of the table. When the lever has passed point B, it slips and the compressed springs let the table fall down abruptly. So that no impact of the lever against the cam occur, at the end of the travel of the table there is put into operation damping device 6, which retards the travel of the table and the movement of the lever. Preliminary compression of the springs may be regulated with the aid of worm gears 7 and 8; there is a dial which allows one to precisely determine the magnitude of the spring compression.

The change in the spring force regulates the acceleration, created by the stand. The magnitude of overload j, naturally, will depend on the force of compressed spring R and the weight of oscillating mass G, which includes the weight of the assembly, the weight of the table and of the rod:

$$j=\frac{R}{G}.$$

It is essential to see to it that the fastening of the assembly to the cable is sufficiently rigid, since only then will the overloads acting on the assembly be equal to the overload acting on the table of the stand.

Tests on the jostling stand simulate the conditions of transportation of an individual assembly or of an assembly, installed in an engine and moving along a highway, a dirt road or the railroad.

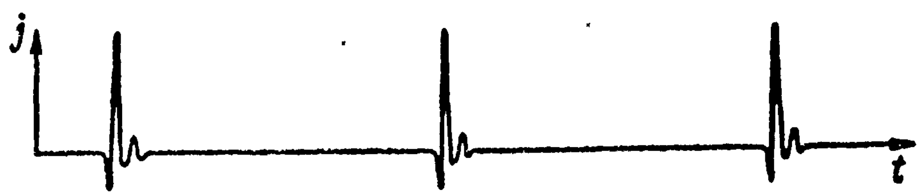

Fig. 6.3. Sample of the type of overloads created by a jostling stand.

A check of the assemblies under the action of a constantly directed acceleration is made in a centrifuge (Fig. 6.5). These tests simulate the operating conditions of the assemblies during accelerated movement of the flight vehicle under the action of the thrust force of the engine.

During the rotation of table 1 of the centrifuge centrifugal acceleration is developed, acting on assembly 8 which is attached to the table. This acceleration is directed along the radius of the table, to the axis of rotation, and will be the greater, the greater the rpm of the table end and the larger the radius of fastening of the assembly r.

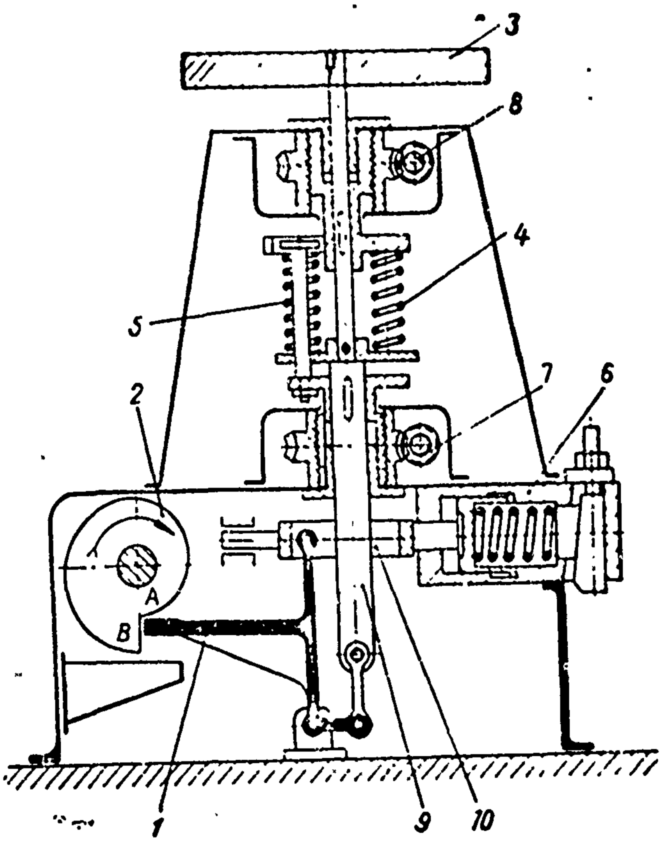

Fig. 6.4. Diagram of the jostling stand: 1 - lever; 2 - cam; 3 - table; 4, 5 - springs; 6 - damper; 7, 8 - worm gears; 9 - rod; 10 - link.

The magnitude of acceleration ε is calculated from the formula

$$\varepsilon=\left(\frac{\pi n}{30}\right)^2 r.$$

Numerically the value of the overload, j = ε/g, is expressed thus:

$$j = \frac{1.01n^2}{900} r.$$

When fastening an assembly of large size close to the center of the table of the centrifuge the centrifugal forces at various points of the assembly will be nonparallel. It is therefore more advantageous to use the periphery of the table; here we will obtain a minimum number of revolutions for a given overload value.

To accomplish triggering of the pneumatic assembly during the action of an overload, beneath the table of the centrifuge there are fastened cylinders, filled with compressed air prior to the tests. From the cylinders air enters the reducer, installed on the table of the centrifuge, and further enters, through the EPV - into the tested assembly (in Fig. 6.5 the EPV serves as the tested assembly).

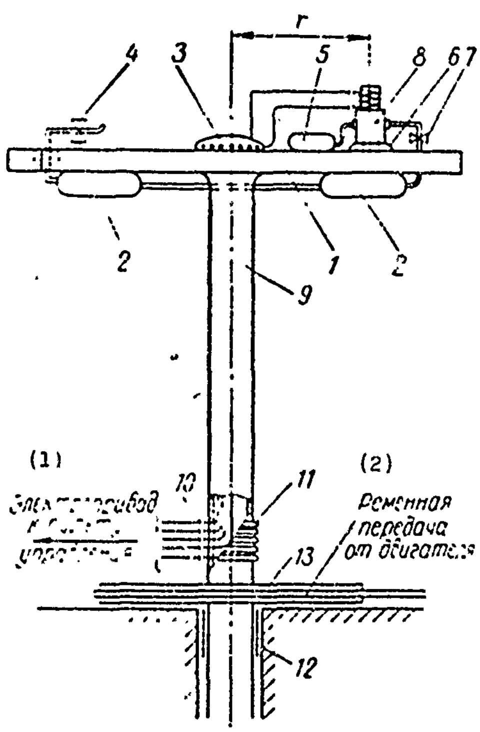

Fig. 6.5. Diagram of the stand of the centrifuge: 1 - table of the centrifuge; 2 - cylinders with compressed gas; 3 - terminal block; 4 - valve for filling with compressed gas; 5 - buffer volume; 6 - bracket; 7 - valve; 8 - tested EPV; 9 - shaft of the centrifuge; 10 - brushes; 11 - circular current collector; 12 - bearing; 13 - pulley.
KEY: (1) Electric lead to control console; (2) Belt drive from motor.

The assemblies are checked for vibration stability on vibration test benches. With the tests on the vibration stands an attempt is made to simulate those accelerations which the assembly receives during the operation of the engine. Practically speaking, this is not always successful, since the possibilities of the test benches for a given weight of an assembly are limited - both with respect to frequency, and with respect to overloads. Therefore, the significant values of frequencies and overloads, sometimes encountered under actual conditions, are compensated for on the vibration stand by the test duration.

Ordinarily the assembly is tested for the effect of acceleration in two mutually perpendicular directions - along the axis of the assembly and perpendicular to it.

Many different types of vibrations stands exist (mechanical, electrodynamic, electromagnetic and others). Figure 6.6 gives a diagram of a very simple, but very reliable mechanical low-frequency (up to 70 Hz) vibration stand, type VP-70.

Electric motor 1 through belt drive 2 imparts rotation by two shafts of vibrator 3, rotating with equal rpm in different directions. On the shafts are mounted disks, on which there are two unbalanced loads 4 each, situated in such a way, that the horizontal components of the centrifugal forces are always mutually balanced. Such a system of loads creates a periodic unbalanced force which is directed vertically. This force varies according to a sinusoidal law. The vibration frequency is regulated by the change in the number of revolutions of the electric motor.

The entire vibrating system is suspended on supporting spring 6. With the aid of a special nut the vibrating system is set up in such a way that the axis of the electric motor and the average position of the shafts of the vibrator are in one horizontal plane.

Vibration amplitude will depend on the weight of the tested assembly and on the position of loads 4; the approach of the loads increases the unbalanced force, and consequently, also the vibration amplitude.

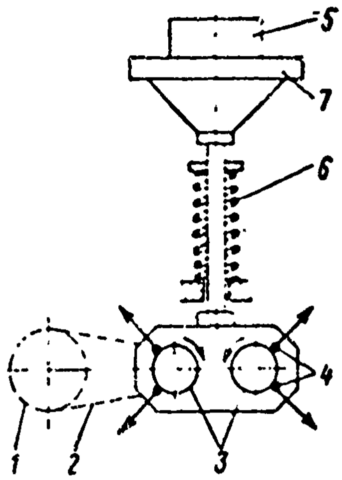

Fig. 6.6. Diagram of the vibration stand VP-70: 1 - electric motor; 2 - belt drive; 3 - vibrator shafts; 4 - loads; 5 - tested assembly; 6 - spring; 7 - table of the vibration stand.

When the assemblies are checked for vibration stability, they must be rigidly fastened to the platform (table) of the stand; if they are not fastened with sufficient rigidity, additional, uncalculable accelerations will arise.

Sometimes, in order to evaluate the working capacity of the assembly's fastening components, the conditions of fastening of the assembly on the engine are simulated on the table of the vibration stand.

The vibration tests are conducted with air pressure supplied to the assembly. The pressure is supplied to this or that cavity of the valve depending on the actual operating conditions. When testing pneumatic assemblies under conditions of vibration several triggerings are made.

The magnitude of the overload j created by the stand with known amplitude (one half the vibration range) A (in mm) and frequency f (in Hz) is determined (independently of the type of stands) from the formula

$$j = \frac{4\pi^2 f^2 A}{9810}.$$

Let us examine the methods of determining the nonhermeticity of seals by checking with compressed gas and let us examine the equipment employed for this purpose. The gas, seeping through the locking mechanism, is discharged through a tube to measure its quantity; in the majority of cases this may also be done while measuring the degree of nonhermeticity of the seals and rings. The quantity of gas in a large discharge can be measured using a rotameter or spirometer; small leaks can be determined by calculating the air bubbles, exiting from a tube per unit of time, with the tube with an inner diameter of 4 mm lowered to a depth of 3-5 mm into alcohol, benzine or water; with more considerable leaks these bubbles are collected into a measuring glass (Fig. 6.7).

It is more difficult to determine nonhermeticity in flanged joints, especially at low temperatures. Sometimes special clamps are made, encompassing the entire perimeter of the connection and insulating it from the environment. The tube exits from the inner cavity of the clamp. However, in the majority of cases, this cannot be done. Then nonhermeticity is determined by soaping down[1] the joint. To determine the hermeticity along the entire perimeter of the joint one can use rotating tables; with stationary tables - one can use a system of mirrors or television units. Frequently the nonhermeticity of joints is determined by the immersion into a liquid - water, benzine, alcohol, freon (the aquarium method), and sometimes - from the pressure drop or, if

[1]If necessary, a special low-freezing neutral soap emulsion is used, based on an antifreeze which is nonfreezing at -50°C.

the connection is not too critical and the requirements imposed on it are not high, - nonhermeticity is determined even aurally.

A more laborious, but on the other hand reliable, method of determining nonhermeticity consists in using haloid or mass-spectrometer leak detectors. A leak detector test is frequently made at low pressure, which is supplied inside the assembly: a freon check (a haloid leak detector) or a helium check (a mass-spectrometer leak detector) with an excess pressure of 2-10 at is approximately equivalent to a test with air at a pressure of 100-200 at.

The mass-spectrometer leak detector is much more popular, although it is significantly more bulky and complex than a haloid leak detector; the freon employed in haloid leak detectors can theoretically interact with the rubber of the parts, causing their softening and swelling[1].

With the absence of rubber parts in the joint the use of haloid leak detectors is also permissible. The principle of their operation consists in the following. Heated platinum emits positive ions; with the content of traces of haloids (freon) in the air the magnitude of the emission increases abruptly; the ion current after amplification is recorded by an instrument. A sensor with a platinum anode is made in the form of a feeler. All the remaining units - the amplifier, the dial instrument, etc. - are located in the housing of the leak detector.

The procedure for checking hermeticity consists in the feeler's inspection of the perimeter of the connection, inside which freon-12,

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[1]With prolonged contact of the parts with freon such swelling actually takes place.

freon-13 or a mixture of a halogen with air is fed. The feeler is slowly moved by hand, at a rate of about 2 mm/sec. When the feeler falls into a place, from which gas is leaking, the emission of ions increases sharply, which can be judged from a sound signal and, quantitatively, from the instrument dial. Thus, the leak indicator establishes both the site and the magnitude of the leak. The sensitivity of the device is very great - it reacts to a concentration of halogens beginning with 0.0001%. This is a disadvantage of it, since the instrument can react to accidental gas leakages, which impedes its use. Very careful ventilation of the location where the instrument is operated is necessary.

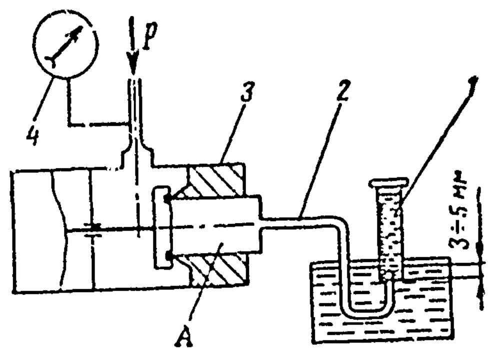

Fig. 6.7. Determination of non-hermeticity of valve seals using a measuring graduate: 1 - graduate; 2 - tube; 3 - tested valve; 4 - manometer; A - cavity behind the tested connection.

The principle of operation of a more perfected, mass-spectrometer leak detector consists in the ionization of the air which surrounds the tested joint, the separation and the recording of the presence of molecules of a certain sample gas. Most frequently the sample gas is helium, and less frequently - argon.

The operating procedure in mass spectrometer and haloid leak detectors is similar. The air surrounding the tested joint is sucked in by the feeler of the leak detector and is then analyzed.

The detection of traces of probe gas (helium) testifies to the nonhermeticity of the tested joint. The degree of nonhermeticity is indicated by the instrument, set up on the panel of the leak indicator. The sensitivity of mass spectrometer leak indicators is even higher than haloid leak indicators.

Leak indicators are described in detail in work [8].

The hermeticity check of a connection by low pressure will sometimes produce not completely true results. Under working conditions the joint will be deformed under the action of high pressure, which can be reflected in its hermeticity. Therefore a more reliable means of checking the hermeticity is a test using working pressure, with the addition of freon or helium to the compressed air. The higher is the partial pressure of the helium (freon) in the mixture, the more rigorous is the check. The mixture composition is regulated and checked using a "flow indicator" which is a specially made d' charge jet with a microscopically small opening, through which the gas is discharged, causing a certain deflection of the instrument's pointer.

A disadvantage of such a method of checking joints under high pressure is the complexity of the organization of carrying out the work with a feeler at a distance, without the presence of people (other methods permit remote observation).

There are several more methods of determining nonhermeticity, chiefly, of flanged joints, but up to this time they have not been very popular. They include: the luminescent method, the methoa of "residual stable deformations" and others.

The luminescent method is based on the ability of certain liquids (luminophors) to be illuminated under the effect of radiation by ultraviolet rays. The assembly is filled with a luminophor (under pressure) and on the outside is irradiated

with ultraviolet rays. Illumination (it remains after the pressure is removed) indicates nonhermeticity.

In monitoring with the "residual stable deformation" method the monitored object is covered with a thin elastic film of natural or synthetic latex, which forms bubbles (or craters - if the bubbles burst) at the sites of nonhermeticity when pressure is fed inside the object.

Test benches for testing assemblies using propellant are rather complex. The purpose of the tests on these stands is to check the effect of the propellant on the state of the mechanical-rubber and plastic parts, and also to check the interaction of the propellant with the employed lubricants, cements, and so forth with the assigned pressure and temperature values. The explosion safety of the design and the hermeticity of the seals on the components are being investigated.

Since the majority of propellants employed for LPREs are either an explosion hazard, or are toxic, or are a fire hazard, or possess all of these properties, it is therefore obvious that the stands must satisfy a number of specific conditions.

The test benches must be made for a certain type of fuel or oxidizer. There are specific requirements for stands with each type of propellant. Let us name only the most common of them.

The test benches must provide the possibility of maintaining the working capacity of the assembly, filled with propellant, for a prolonged period in the entire operating temperature range. Therefore the test benches must possess a device for automatic maintenance of the given temperature.

As a result of the corrosivity and toxicity of propellants the control of the entire test process, excluding the assembly

and disassembly of the unit, must be remote, beginning with the filling of the propellant into the test bench tanks and ending with its drainage after the operation is finished; after the assembly is taken down (after the tests) it, along with the tubing connected to it, must be thoroughly blown out with a neutral gas in order to remove any propellant residues.

The measurement of leaks of the product (nonhermeticity of the seals of the assembly) should be made with the aid of chemical reagents (absorbents), connected to the drainage outlets. The hermeticity of the external joints is checked with indicator bands.

The test bench tanks must be equipped with remote level indicators, showing the quantity of the product. The premature emptying of the working tank or the overfilling of the drainage volume can lead to serious consequences.

The stand should be equipped with intake and exhaust fans, and at the working location there should be a signal indicator, showing the gas concentration of the atmosphere (a recording gas analyzer).

The test bench volumes must be equipped with protective and normally-closed drainage[1] valves, preventing the outflow of propellant vapors to the atmosphere and the entry of moisture inside the tanks.

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[1]With low-boiling propellants the drainage valves must be normally-open valves, in order to prevent an inadmissible pressure increase.

All the drainage and ventilator discharges must not go directly into the atmosphere, but must be first neutralized in special systems[1]. Similarly, liquid propellant, which has fallen onto the floor of the test bench, must not be drained into the ordinary sewer system.

In case a fire breaks out, the propellant tanks must be immediately emptied (there must be an emergency discharge system); the propellant must be emptied into a special tank, some distance removed from the test bench. A fire-extinguishing system must be provided for, in certain cases to include the filling of the test bench with an inert gas, preventing the spread of flames.

A special place among the various types of tests is occupied by tests which take the hydraulic characteristics of propellant valves, which have important significance for determining the working capacity of an assembly.

The following section is dedicated to this problem.

6.3. EXPERIMENTAL DETERMINATION OF THE HYDRAULIC CHARACTERISTICS OF PROPELLANT VALVES

Each type of propellant valve during its design-development period, and in a number of instances also each sample of the valve during its manufacture[2], is subjected to water flow testing on a special hydraulic test bench. The purpose of the these flow tests is: to determine the hydraulic resistance of the valve

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[1]The calculation of the quantity of drainage discharges is given in the appendix.

[2]In those cases where the variation in the values of the hydraulic resistance of the valve have a significant effect on the engine characteristics.

under characteristic conditions - under conditions of main-stage, pre-stage, and sometimes also terminal-stage operation; to determine the flow rate of the liquid and the pressure differential, at which the movement of the moving system occurs; and to obtain the dependences of the pressure drop at the valve on the flow rate.

In the development of the valve's design, as a result of these tests a solution can be made on the application of these or other changes in the configuration of the flow-through tract, on the change in the disk profile, type of seat and so forth, i.e., the tests clarify peculiarities of design, not taken into consideration by the calculations.

The basic diagram of a very simple test-flow bench with pump supply is shown in Fig. 6.8. Water from the service tank is supplied by a centrifugal pump with an electric-motor drive. The water flow rate and pressure are regulated by cutoff valves 1 or 2; the flow rate value is measured by a flow rate meter of one type or another (for example, a narrowing device with a differential manometer; an electromagnetic flow rate meter; weights and so forth). The initial pressure in front of valve 1 is determined first by the operating conditions (rpm) of the pump.

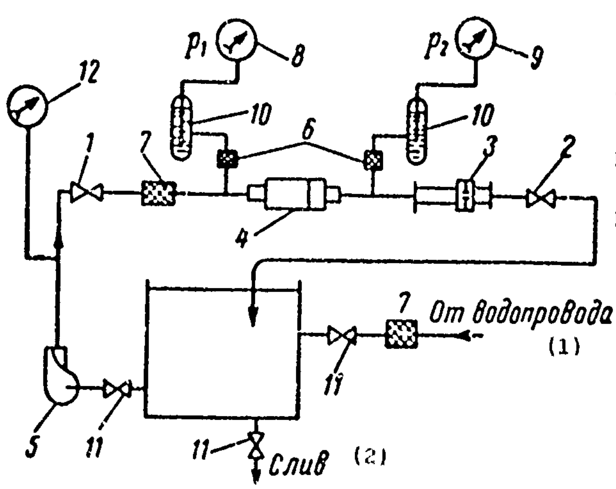

Fig. 6.8. Diagram of a flow bench with pump feed: 1 - inlet valve; 2 - outlet valve; 3 - flow rate meter; 4 - tested assembly; 5 - pump; 6, 7 - filters; 8, 9 - manometers; 10 - dampers; 11 - stop valves; 12 - manometer.
KEY: (1) From the water supply; (2) Drain off.

The water passes through the tested valve, and returns to the tank.

At low flow rates the valve's characteristics can be taken on the hydraulic test bench with a pressurized feed system, a diagram of which is shown in Fig. 6.9. Here the initial pressure in front of the valve is determined by the magnitude of the setting of the air pressure reducer 4, while the flow rate is regulated by sluice valve 8.

The pressure differential Δp on the valve is determined as the difference in static pressures at the inlet and outlet: $\Delta p = p_1 - p_2$. During the flow tests the pressure p_2 at the outlet from the assembly[1] is usually maintained approximately constant with the aid of a sluice valve (for example, in Fig. 6.8 - sluice valve 2) and, by varying flow rate Q by the turning of inlet valve 1, the dependence of the static differential Δp on the flow rate is obtained[2].

From the flow test results graphs of $\Delta p = f(Q)$ are plotted for the reduction of several measured values of the drop in $\Delta p_{зам}$ to nominal flow rate $Q_{ном}$ are made:

$$\Delta p_{прив} = \Delta p_{зам}\left(\frac{Q_{ном}}{Q_{зам}}\right)^2.$$

This is ordinarily done with low divergences of $Q_{зам}$ from $Q_{ном}$. Nominal value of $\Delta p_{ном}$ is calculated as the average of an obtained reduced values of $\Delta p_{прив}$:

$$\Delta p_{ном} = \frac{1}{n}\sum_{i=1}^{i=n} \Delta p_{i\,прив}.$$

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[1]In the case where the flow tests are conducted under main-stage conditions; in flow tests on pre-stage mode p_2 must not be approximately constant, but strictly constant.

[2]During the water flow tests the calculations use the volumetric flow rate Q, and not the mass flow rate G.

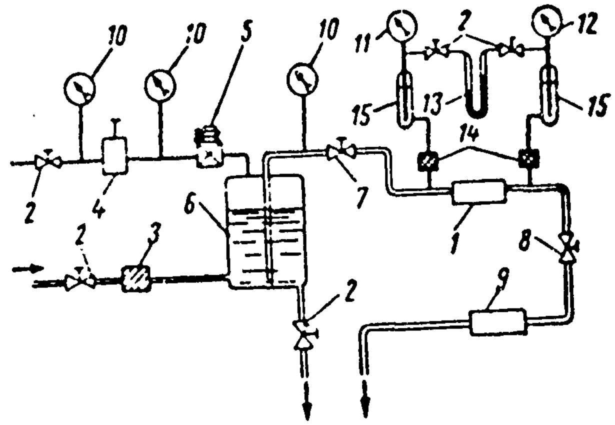

Fig. 6.9. Diagram of a flow bench with a pressurization feed system: 1 - tested valve; 2 - stop valves; 3 - filter; 4 - air pressure reducer; 5 - EPV; 6 - service tank; 7 - inlet valve; 8 - outlet valve; 9 - flow rate meter; 10, 11, 12 - manometers; 13 - differential manometer; 14 - filters; 15 - dampers.

The typical form of a graph of $\Delta p = f(Q)$ on main-stage mode of engine operation is shown in Fig. 6.10. This graph is a parabolic curve of the type $\Delta p = aQ^2$, the multiple a of which is determined by the coefficient of hydraulic resistance ξ:

$$a=\xi \frac{\gamma}{2gF^2},$$

where F is the area of the decisive cross section of the valve. (The outlet or inlet valve cross section is selected as the decisive cross section).

The coefficient ξ within a broad range of change in flow rate Q usually remains approximately constant, in connection with which the Reynolds numbers (Re) in the flow are rather great, while with high values of Re the coefficient ξ almost does not depend on Re (an automodel mode).

Since in main-stage mode the propellant valve is always completely open, while the moving system is pressed with a considerable force to the stop, flow may occur under a pressure significantly less than working pressure. In view of the low compressibility of water, the magnitude of the drop is determined only by the flow rate, and not by the pressure. The possibility of flow under low pressures is very important, since at high flow rates the test bench pumps do not always permit the assurance of high working pressures. If the pump does not provide a sufficiently high pressure head, before the flow test of a normally-closed valve the spring is removed from it in order to reduce the working pressure. This can be done only in the case where it is clear that the locking mechanism cannot move during working mode.

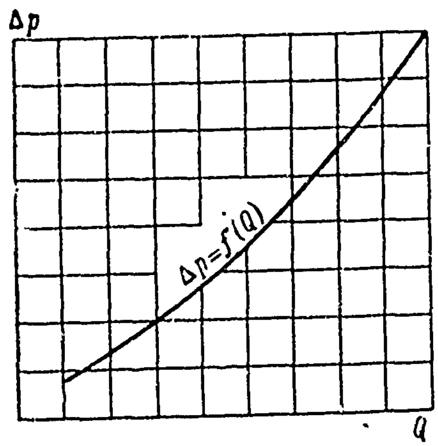

Fig. 6.10. Dependence of the pressure differential Δp on flow rate Q during operation on main stage mode.

To conduct flow tests at reduced pressures it is essential to first be convinced that no cavitation will occur, since this will limit the pressure reduction. The presence of cavitation may lead to significant errors in the test results. Therefore the determination of the hydraulic characteristics of the valves (just as the flow tests of the discharge jets, throttles and other engine components) should begin with the taking of the cavitation characteristics $\Delta p = f(p_2)$, i.e., with the clarification of the effect of counterpressure p_2 on the magnitude of Δp. The

taking of the cavitation characteristics is done during the flow tests of the valve at a constant flow rate (equal to the nominal flow rate or exceeding it), determining the pressure differential Δp with the change in counterpressure p_2. Under cavitationless conditions the value of p_2 should not influence Δp.

The typical form of a cavitation characteristic curve is represented in Fig. 6.11. It is evident from this that, beginning with some value $p_{2доп}$ a further decrease in the counterpressure produces a rise in Δp. During the flow on the bench it is desirable that the value of p_2 exceed the value $p_{2доп}$ slightly. The greater the flow rate (the greater the speed of the liquid), the more abruptly occurs the phenomenon of cavitation, and the more abruptly the value $p_{2доп}$ increases.

To obtain accurate results the flow tests of the assemblies must be organized with careful observation of the recommendations cited in "Rules 28-64 for the measurement of the flow rate of liquids, gases and vapors with standard diaphragms and nozzles"[1] and in the book of V. P. Preobrazhenskiy "Technological measurements and instruments"[2]. Although these recommendations (relative to the lengths of the straight portions, the location and connection of the measuring instruments and so forth) pertain to flow meter instruments, nevertheless, as experiment has shown, they must also be used for the flow testing of valves. A deviation from the cited rules may result in the flow tests of one and the same assembly showing different results using different equipment. The reason for this is found in the different configurations of the flow and in the system of organization of pressure measurement.

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[1]Publishing House of the Committee of Standards, 1964.
[2]Gosenergoizdat, 1953.

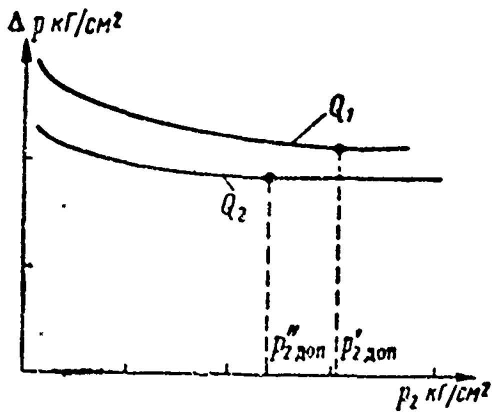

Fig. 6.11. Cavitation characteristic curve (dependence of the pressure differential Δp on the counterpressure p_2 with an unchanging flow rate): $Q_1 > Q_2$.

The flow testing of valves in prestage mode in the area of the start of opening to main stage must be done under working pressures. This requirement is caused by the fact that the opening (closing) force of the valve is determined not only by the pressure differential Δp, but also by the propellant pressure itself (see, for example, Fig. 2.3). Therefore the flow tests at nonworking pressures p_2 even at an assigned flow rate value, lead to errors in the determination of Δp - in view of the obtained nonconformity of the opening of the valve with the true opening (nonconformity of the flow passage cross-sectional areas).

Figure 6.12 shows the characteristic graph of the flow test of a valve (see Fig. 2.10) working on prestage mode. This graph up to point A is analogous to the curve shown in Fig. 6.10, i.e., is a quadratic curve; in this section the unchanged position of the valve disk is preserved. At flow rates which exceed Q_A, the valve begins to move, opening for operation on main-stage mode, and the quadratic characteristic curve is disturbed.

The value Δp, corresponding to the operation of propellant valves on prestage mode, can be affected by various, and at first glance insignificant, factors. Thus, the tightening of the nuts on a flange of the inlet tubing (see Fig. 2.10) can lead to the

deformation of the valve body, which is reflected in the magnitude of the valve's flow passage cross sectional area during its operation on prestage mode, which is small in absolute value; as a result, it turns out that the change in the degree of tightening of the nuts does affect the value of Δp (Fig. 6.13).

Figure 6.14 shows the effect on the value of Δp of the valve's service life. Curve 1 (after nominal assigned service life) lies below curve 2 since, as a result of the valve's triggerings the rubber seal is deformed, as a consequence of which the flow passage cross sectional area for the liquid is increased and the value of Δp decreases.

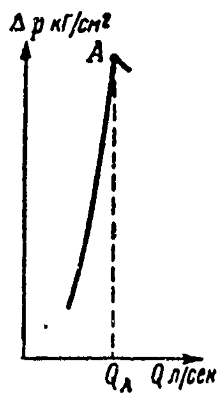

Fig. 6.12. Hydraulic characteristic Δp = f(Q) on prestage mode (at point A - the start of movement of the moving system).

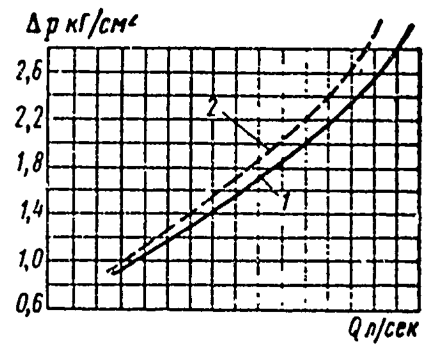

Fig. 6.13. The effect of the degree of tightening of the nuts of the flange of the inlet tubing on the value of Δp: 1 - initial tightening; 2 - tightening of the nuts is increased by three facets.

If pressure p_2 at the outlet from the assembly is gradually increased, the valve begins to open for operation on main stage mode and will be capable of passing (at the same or at a lower pressure on the inlet, or, what is the same - with a constant or with a smaller pressure differential Δp) a large flow rate.

The characteristic curve, shown in Fig. 6.12, is continued into the region of increase in the flow rate and is represented in Fig. 6.15. We should note that during engine operation an increase in pressure p_2 occurs as a result of the pressurization in the combustion chamber, and an increase in the flow rate - as a result of the increase in the number of rpm of the TPA; but on the test bench an increase in p_2 may be achieved by different means: by an increase in the rpm of the pump with an unchanged position of sluice valve 2 (see Fig. 6.8), by the closing of sluice valve 1 with unchanged rpm of the pump, by the partial closing of sluice valve 2 or by the combined effect on the sluice valves and on the pump. In all these instances the change in p_2 is related to the change in the flow rate. Therefore, with the opening of the valve (in the area of flow rates which exceed Q_A) the curve $\Delta p = f(Q)$ characterizes not the valve, but the conditions of the flow test (see Fig. 6.15). Here the dashed lines 1-5 indicate the possible changes of the dependence of $\Delta p = f(Q)$ with the shifting of the locking mechanism, beginning at point A; here the flow passage cross-sectional areas in the valve increased according to various laws, determined by the method of action on the mode.

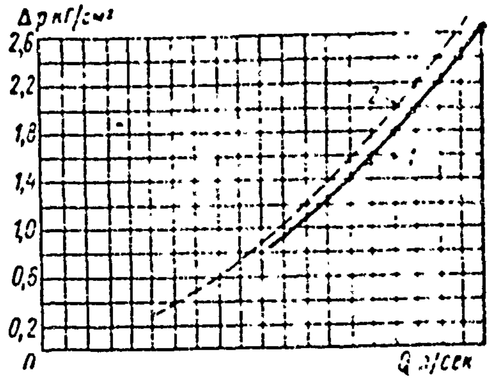

Fig. 6.14. The effect of operating service life on the value Δp: 1 - after assigned nominal service life; 2 - up to assigned nominal service life.

If with the increase in the flow rate p_2 is artificially maintained constant, then the opening of the valve will occur only as a result of an increase in Δp - the pressure differential on the valve disk; the rise in Δp may occur only as a result of

the increase in the speed of the liquid in the throttling slot between the seat and housing 1 (see Fig. 2.10) and valve disk 3. Dashed line 1 in Fig. 6.15 shows the character of dependence $\Delta p = f(Q)$ in this case. The slope of straight line 1 characterizes the rigidity of the springs, i.e., indicates the necessary increase in the pressure differential for their compression. (This is correct, naturally, only in the case where the losses at the inlet to the assembly and at the outlet from it are negligibly small in comparison with the pressure drop on the valve disk; ordinarily this is what occurs).

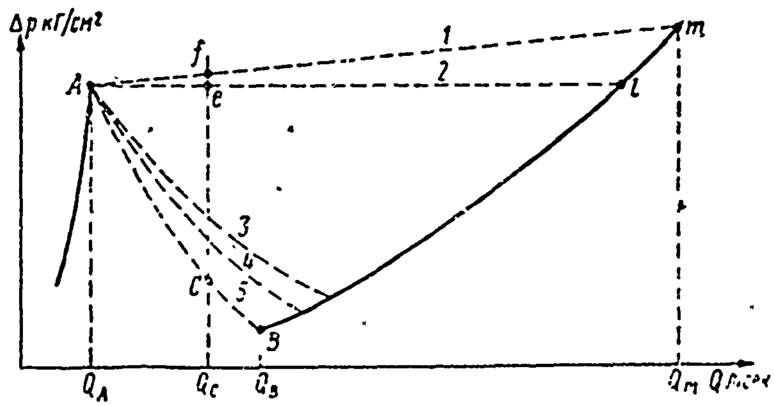

Fig. 6.15. Characteristic curve of $\Delta p = f(Q)$ with the closing of the valve.

If the sluice valves (see Fig. 6.8) are manipulated, so that with the increase in the flow rate the drop in Δp remains constant, i.e., a constant velocity is maintained in the slot between the seat and the disk, then the opening of the valve will occur only as a result of the rise in p_2 (curve 2 in Fig. 6.15). The dashed curves 3, 4, and 5 demonstrate the nature of the opening of the valve with various laws of increase in p_2. The value of Δp drops as a result of a decrease in the liquid velocity in the slot with a large valve opening; with one and the same flow rate Q_c the opening (travel) of the valve h will be different: the greatest opening is at point C, the smallest - at point f, where there is the least pressure p_2. The most abrupt rise in p_2 occurs along curve 5.

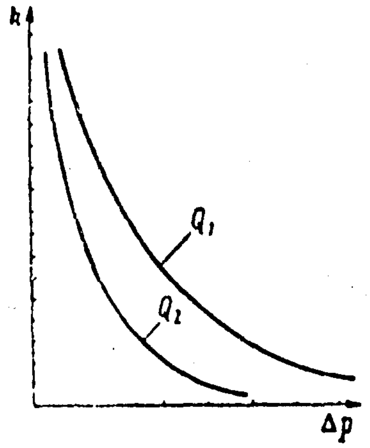

Fig. 6.16. The dependence $\Delta p = f(h)$.

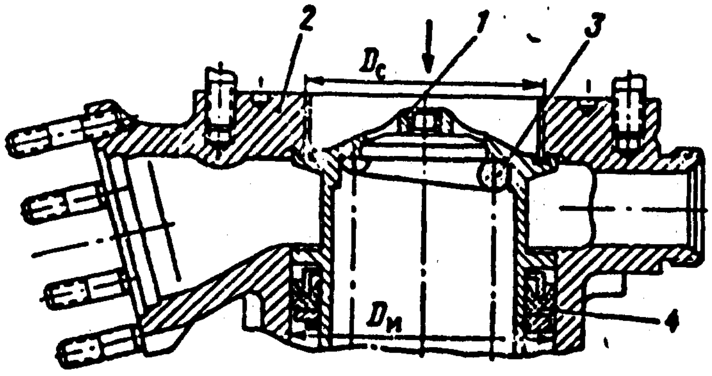

Fig. 6.17. Sketch of a normally covered (partially open) valve of the discharge type: 1 - valve; 2 - body; 3 - spring; 4 - seal ring.

If the valve opened completely for operation on main stage mode with minimum flow rate Q_B, then the value Δp is determined by point B, where the pressure on the outlet p_2 will be least; but at pressure p_2, equal to the pressure on prestage mode (with the opening of the valve according to the law p = const) the valve opens completely only with significantly greater flow rate Q_m.

After the complete opening of the valve for operation on main stage mode the graph $\Delta p = f(Q)$, independently of the opening principle, will again represent a quadratic dependence.

In Fig. 6.15 the curve Bm represents the dependence $\Delta p = f(Q)$ when $p_2 = p_{2max}$.

For studying the actual character of valve opening it is necessary to know the principles relating the change in the propellant flow rate with the pressure behind the valve. However, ordinarily for a new engine they are unknown. Therefore, in

studying a valve's operation the dependences $\Delta p = f(h)$ are assumed when Q = const (curves obtained with various Q, see in Fig. 6.16) or the dependences $\Delta p = f(Q)$ are assumed with various values of h (constant for each test).

As an example, let us examine the characteristics of the process of opening of a normally covered (partially open) valve of the discharge type, shown in Fig. 6.17. The characteristics were taken on a flow bench with a pump feed.

If there is no pressure in the controlling cavity then the process of valve opening begins with the achievement of a liquid pressure equal to p_1' at the valve inlet (Fig. 6.18). At this moment, when there is still no flow, the counterpressure at the outlet from the valve p_2 is, naturally, equal to zero. Subsequently with the opening of the valve the flow of liquid begins and counterpressure p_2 appears. An increase in the counterpressure favors the opening of the valve - outer diameter D_M (see Fig. 6.17) of seal ring 4 exceeds seat diameter D_c. The opening of the valve with the increase in pressure and flow rate of the liquid continues with the increase in the pressure differential Δp on the disk (see Fig. 6.18, heavy line).

With total opening, when the valve has reached the detent in the body, the pressures at the inlet and outlet are almost equal, and the pressure differential Δp becomes minimal (see Fig. 6.18, point A). With the further increase in the flow rate the differential increase will occur according to the usual quadratic curve $\Delta p = aQ^2$.

The curve of the pressure differential $\Delta p = f(Q)$ on the segment to the right of point A (see Fig. 6.18) has a stable character. It is stable for all assemblies of the same construction, although curves obtained for various samples may differ slightly from one another due to the variation in values of ξ.

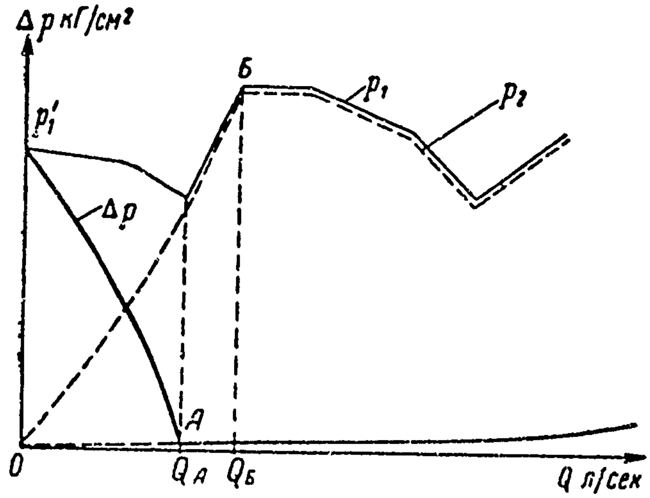

Fig. 6.18. The change in Δp with the increase in the flow rate Q (point A - the moment of complete valve opening; to the right of point Б - arbitrary regulation of the outlet pressure).

The curve of the pressure differential $\Delta p = f(Q)$ with flow rate less than Q_A (initial opening period) has an unstable, unsteady character. The position itself of point A depends on the spring force and the counterpressure value. For normal operation it is required that the initial valve opening period occur very quickly in time. It is important that the nominal flow rate through the valve during operation on main stage mode be considerably to the right of point A.

Figure 6.18 shows the values of the pressure at the inlet p_1 and at the outlet p_2, obtained during bench flow tests of the assembly at relatively low consumption rates. The value of the counterpressure at flow rates greater than $Q_Б$ was manually regulated on the test bench by valves. At flow rates less than $Q_Б$ regulation of the position of the test bench valves was not done.

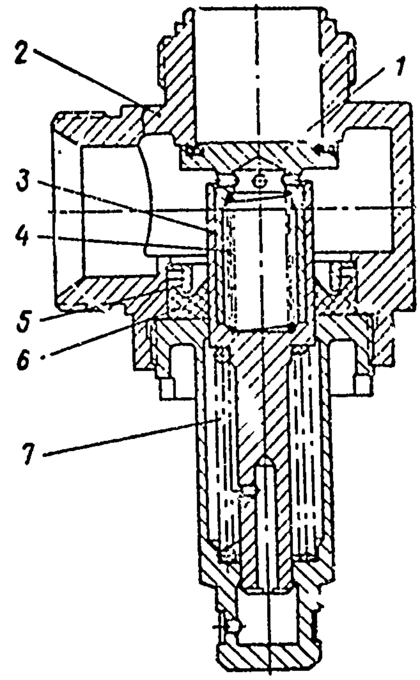

Fig. 6.19. Special relief valve: 1 - valve; 2 - body; 3 - bushing; 4 - auxiliary spring; 5 - collar; 6 - seal ring; 7 - main spring.

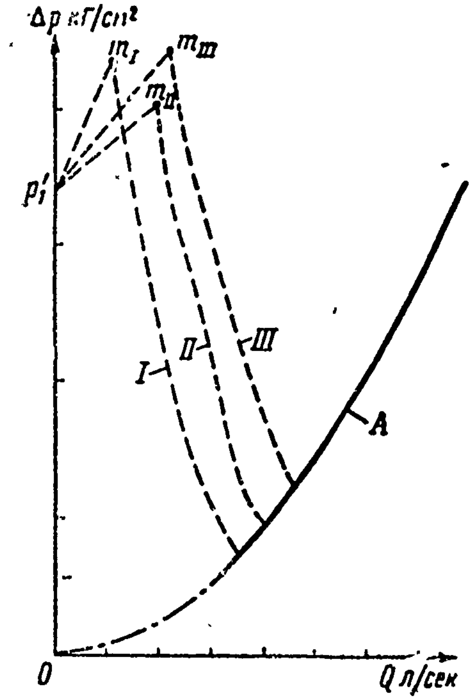

Fig. 6.20. Characteristics of the valve opening: p_1' - pressure of the start of the valve opening.

Let us examine anoth r example. The operation of the valve of a gas generator, depicted in Fig. 6.19, is clear from the figure. This is a repeat-action valve but it is not pneumatically controlled; closing is provided by spring 7, and opening - by th propellant pressure. Its operation is similar to the operation of the valve shown in Fig. 6.17. In essence - this is a special-purpose relief valve. Depending on the value of the pressure p_2 at the outlet from the valve, the valve opening process will have a different character.

Figure 6.20 contains curves of $\Delta p = f(Q)$ for one and the same valve sample with various loads[1] at the outlet. With a small counterpressure and a low flow rate the value of Δp is not subject to the quadratic law (curves I, II, III), since the valve is in the process of movement, and with fixed p_2 the nature of its opening is determined by its rigidity of spring 7 (see Fig. 6.19) (the rigidity of spring 4 is not taken into consideration, since valve 1 will rest in the end of bushing 3). Only then, when the valve is completely opened (and this occurs with an increase in the pressure differential as a result of the increase in the flow rate), the graph of $\Delta p = f(Q)$ becomes quadratic (see curve A in Fig. 6.20). With different loads this will correspond to different flow rates.

As can be seen from Fig. 6.20, various values of Δp can correspond to one and the same flow rate value, depending on the counterpressure p_2. With the increase in the counterpressure the points of intersection of curves I, II, III and curve A will approach the origin[2]. The loading of the system in the case of curve I is higher than in the case of curve III. It is characteristic that the "peak" values of Δp_m of curves I, II, III are nonuniform: thus, the peak value of curve II is less than the peak values of curves I and III, although the system load is average for it. This speaks of instability, of indeterminacy of the valve's operation at low flow rates.

To ensure stability of the processes it is necessary that the curve $\Delta p = f(Q)$ have a definite character in the large range of flow rates, i.e., that the intersection by a curve of type I and by curve A occur at flow rates as small as possible; it is

[1] From here on in the book by load we mean the system of tubing, including valve 2 (see Fig. 6.8), which creates the hydraulic resistance that determines the value of pressure p_2.

[2] Each curve was taken at a constant loading at the outlet, but the counterpressure p_2 during the process of the flow tests increased with a rise in the flow rate.

unconditionally necessary that the nominal mode be located on a stable segment of the characteristic curve, i.e., on curve A. With a sufficiently large counterpressure, determined by the force of spring 7 (see Fig. 6.19), the graph of $\Delta p = f(Q)$ almost throughout conforms to the quadratic law. From this viewpoint it is desirable that spring 7 be extremely weak. However, in practice, this cannot be allowed since the valve's purpose is to hold back the start of the component flow until reaching a fixed pressure at the inlet, equal to p_1' (see Fig. 6.20).

A somewhat different picture will occur in flow testing ordinary relief valves. The magnitude of the pressure drop Δp will in this case not depend on the value of the counterpressure. An example of such an assembly may be the valve depicted in Fig. 6.21. The results of flow testing it are shown in Fig. 6.22. With a pressure at the inlet, equal to p_1', valve 2 (see Fig. 6.21) begins to open. However, with an increase in the flow rate the value of Δp does not fall, as occurs in Fig. 6.18, but usually increases (see Fig. 6.22). The character of the change in Δp is in this case determined basically by the rigidity of spring 4 (see Fig. 6.21) and by the hydrodynamic forces. During the opening spring 4 is compressed and the increase in the pressure differential Δp compensates for the increase in the compression force of the spring. Similarly Δp increases (see Fig. 6.20) during the opening of the valve, depicted in Fig. 6.19.

The maximum value of Δp (Δp_{max}) (see Fig. 6.22) occurs at a certain value of the inlet pressure p_1, which we shall call the "reversal pressure". The value of the "reversal pressure" p_1 practically coincides with the value of Δp_{max}, since p_2 is at this moment very small.

Complete opening of the valve commences with a flow rate equal to Q_L, and at this moment valve 2 (see Fig. 6.21) reaches its detent in body 5; only after this does the value of Δp begin

to fall. The subsequent force of curve $\Delta p = f(Q)$ does not depend on the spring compression force. With a flow rate greater than $Q_Б$ the characteristic is stable and steady.

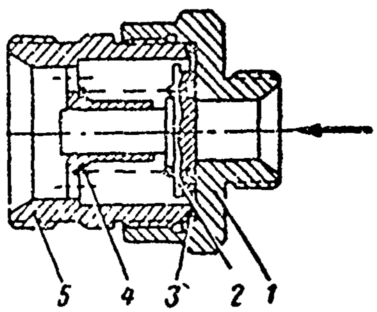

Fig. 6.21. Relief valve: 1 - connector; 2 - valve; 3 - gasket; 4 - spring; 5 - body.

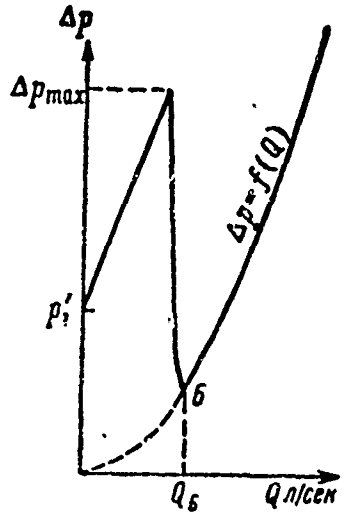

Fig. 6.22. The hydraulic characteristic of the relief valve: $Q_Б$ flow rate at which the valve is completely opened.

Curve $\Delta p = f(Q)$, depicted in Fig. 6.22 for the given valve sample, as was stated above, is in principle designed for any counterpressure. However, for another valve sample it may have a somewhat different character and the magnitudes of Δp will have different values. At low flow rates, when valve 2 (see Fig. 6.21) hangs on spring 4 and has not yet reached its detent in body 5, the magnitude of the pressure differential will depend on the individual peculiarities of the spring. The greater the spring force, the greater must be the flow rate of liquid in order for valve 2 to reach the detent in body 5, when the opening becomes constant.

In a similar assembly design, at flow rates less than $Q_{\text{Б}}$, pressure fluctuations at the valve inlet and outlet are very probable, and the average value of the hydraulic resistance at these flow rates will be unwarrantedly great. When the valve is completely open, the pressure differential between the inlet and outlet (at a low flow rate) will be small, and therefore the spring will slightly close the valve. With the closing of the valve the pressure at the inlet to the valve increases and the pressure at the outlet decreases; therefore, the pressure differential increases and the valve begins to open.

The rate of the fall and rise in pressure before and after the valve is inseparately connected with the capacity and rigidity of the system of the valve's inlet and outlet. In the case where flexible conduits are employed (bellows, rubber hoses) the system will be subjected to the emergence of auto-oscillations, which cause noticeable pulsations in the pressure and flow rate. With rigid short systems pulsations are less noticeable.

With flow rate $Q_{\text{Б}}$ (see Fig. 6.22) the valve is completely opened and, beginning with point Б, the graph represents a quadratic dependence. Pressure pulsations cease and the regime becomes steady. Nominal valve operating mode must lie within the limits of a quadratic, stable branch of the characteristic curve.

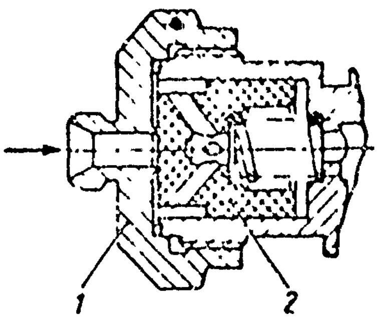

Fig. 6.23. Initial design of a relief valve: 1 - body, 2 - valve.

However, under starting conditions the initial branch of the characteristic curve may also be of great interest. Therefore, let us stop on the question of reducing the "reversal pressure" p_1 and the peak value of Δp_{max} at low flow rates.

Figure 6.23 shows the initial construction of one of these valves. In the slot between body 1 and valve face 2, when the valve opening was small, the liquid flowed at high velocity and therefore the pressure in the slot was reduced; the pressure in the cavity behind the valve pressed valve 2 to its seat and created as it were a "reverse" pressure differential. The presence of this "reverse" pressure differential is equivalent to an increase in the spring force and therefore this design has an inherently large Δp_{max} and a large "reversal pressure."

To reduce the value of Δp_{max} and to decrease the effect of the hydrodynamic forces during flow around the valve, the design of the assembly was perfected, as shown in Fig. 6.24a: a chamfer was made in valve 2. The radial expansion of the slot, where a region of low pressure arises, was significantly reduced, i.e., the stress from the "reverse pressure differential" was reduced. As a result, the region of stable characteristics was increased due to the decrease in the flow rate value $Q_Б$; the "reversal pressure" went down and the value of Δp_m fell. Along with this, the tendency of the valve toward auto-oscillations, and toward pressure fluctuations was reduced.

In another instance, when the height of the valve disk was small and because of design considerations chamfering was impossible, acceptable hydraulic characteristics were obtained by inserting bore holes 3 and by subfacing the disk at a diameter, which exceeded the diameter of the seat (see Fig. 6.24b). As a result, the pressures on both sides of the disk were equalized, and the hydrodynamic forces no longer pressed the disk to the seat, as was done before the modification.

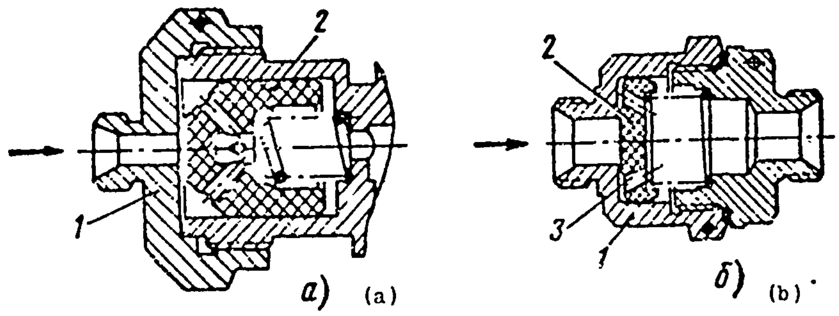

Fig. 6.24. Modified version of a relief valve: a - chamfering; b - face of valve is machined; 1 - body; 2 - valve; 3 - bore.

Figure 6.25 shows still another method of ensuring faster valve opening. The initial design of the valve was similar to the construction shown in Fig. 6.21. In the modification (conforming to Fig 6.25), besides subfacing the disk, the diameter of the disk collar was enlarged, as a result of which the diametric clearance between the valve and the body was significantly reduced. This led to an increase in the force acting on the disk with a small opening. Even with a low flow rate the valve will try to open completely.

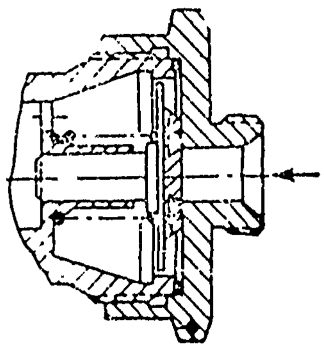

Fig. 6.25. Modification of the valve's design to ensure its sudden opening.

Problems of hydrodynamics played a significant role not only at low flow rates. The nature of the flow around the valve disks by the liquid has a significant effect on the magnitude of the hydraulic resistance under steady state conditions.

Let us examine the data on the flow testing of a normally closed valve, depicted in Fig. 2.7. In this assembly the hermeticity when the valve is seating itself is ensured by the use of a rubber seal. Due to the "adhesion" of the rubber to the seat the work was carried out with the replacement of the rubber seal by a fluoroplastic. Several versions for the attachment of the fluoroplastic seal were investigated, i.e., several variations of the design of valve 3 (Fig. 6.26) with one and the same seat profile were examined. The lower the flow rate and the lower the hydraulic resistance Δp, at which the relief valve opens completely, the better are the valve's characteristics: minimum values of Q and Δp testified to the optimum flow-around process, i.e., to the most advantageous hydraulic characteristics of the valve.

Table 6.1 shows data which characterize the hydraulic resistance of the assembly with different configurations of valve 3 and fairing 4. The perimeters of the valve, depicted in Fig. 2.7, are also shown there for the sake of comparison.

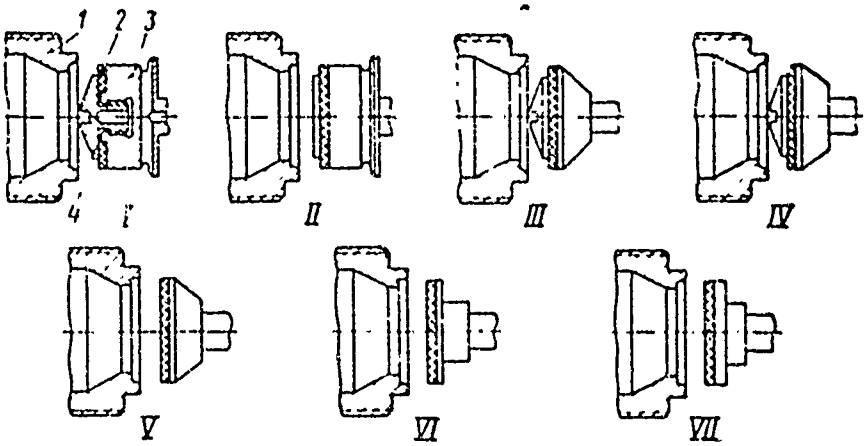

Fig. 6.26. Different variants of the profile of the flow through section of a propellant valve: 1 - seat; 2 - seal; 3 - valve body; 4 - fairing.

Table 6.1.

Number of variant (see Fig. 6.26)	Flow rate value (in l/sec), at which the valve is completely opened	Value of Δp (in kgf/cm^2), at which the valve is completely opened
I*	10.2	16.42
II*	10.15	7.54
III	8.98	4.05
IV	6.47	2.2
V	3.74	0.7
VI	3.91	0.8
VII	7.1	2.55
Valve, shown in Fig. 2.7.	4.03	0.9

*With the cited parameters the valve is still not completely opened.

As can be seen from the data introduced in Table 6.1, the shortening of the cylindrical portion of the valve's disk and the rejection of the conical fairing, which pressed the fluoroplastic seal, exerted a favorable effect on the hydraulic characteristics; a reduction in the diameter of the annular collar of the disk also had a positive effect on the characteristics. However, the most successful (with respect to hydraulics) variants V and VI (see Fig 6.26) require a change in the attachment design of the fluoroplastic seal to the valve.

From the examples given it follows that the factors which influence the process of flow around the valve - the form of the disk, the configuration of the valve's flow-through section, the speed of the flow - are of important significance for the magnitude of the hydraulic losses and the character of the valve's opening.

The stability of the hydraulic characteristics of the assembly is also influenced by the rigidity of the parts located in the flow of the propellant. Let us confirm this with the characteristics of the pneumatic valve, shown in Fig. 6.27. With flow rates in the region Q_A-Q_B (see Fig. 6.28) in certain samples of assemblies there was a jump, a discontinuity of the characteristic curve. This jump occurred in various samples of the valves at flow rates which were somewhat different from one another. During the flow test with an increase in the flow rate (curve 1 in Fig. 6.28) within the region Q_A-$Q_Б$ there was a sharp reduction in Δp; in the flow test with a reduction in the flow rate (curve 2) a rise in Δp was observed (ordinarily with a somewhat lower flow rate); both before and after the jump the change in Δp with a rise in the flow rate had a quadratic character. In the majority of the assemblies no jump in the characteristics was observed, and curves Δp = f(Q) of these assemblies corresponded to the lower branches of curve 1 without discontinuity (see the broken curve in Fig. 6.28).

A special study was conducted for the reasons of the emergence of this jump. As a result of the investigation, it was clarified that the discontinuity of the characteristic curve occurred as a result of the deformation of seal ring 1 (see Fig. 6.27), leading to a change in the profile of the channel for the flow of liquid. With a change in the flow rate deformation of the blade of the seal ring occurred, and with any value of the flow rate the seal ring was reversed, thereby reducing the hydraulic resistance here.

But the majority of seal rings at maximum flow rates, which might have been provided by the test bench equipment, were not reversed (turned inside out), and in these valves no discontinuity of the characteristics occurred.

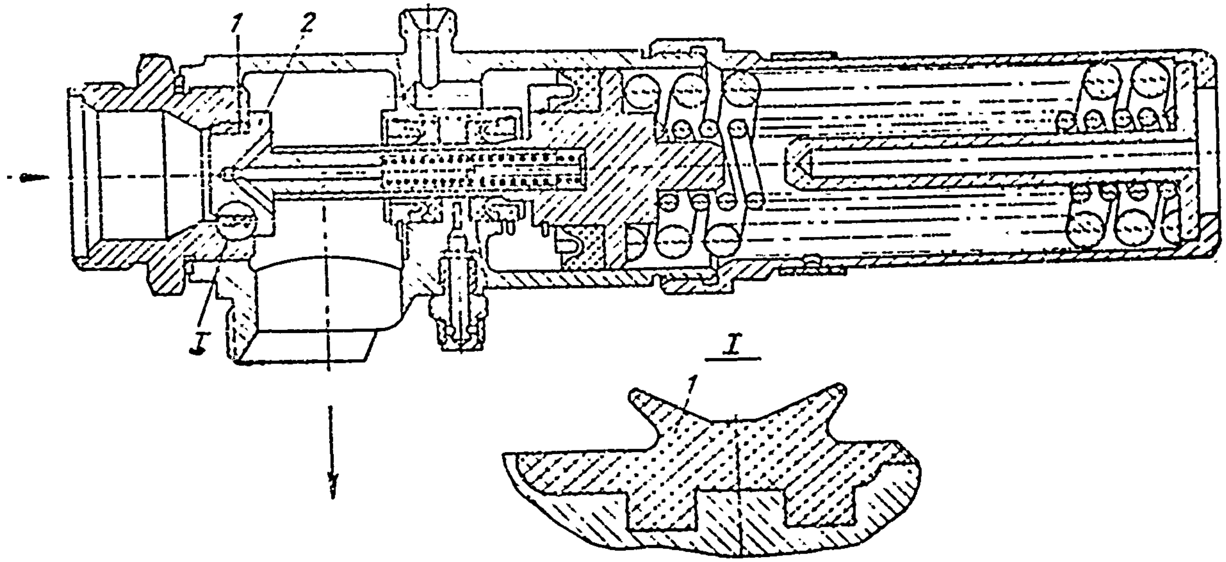

Fig. 6.27 Pneumatic propellant valve with special seal ring (variant of the valve shown in Fig. 2.7): 1 - seal ring; 2 - valve.

The flow rate value at which the reversal of the blade of the seal ring takes place depends on a number of factors (rigidity of the rubber, actual sizes of the seal ring itself and of the mated parts, degree of aging of the rubber, temperature and so forth).

Thus, the presence of a rubber seal in the flow leads to instability of the valve's operation, and consequently, to the instability of the engine's operation as well.

The determination of the hydraulic resistance of valves is made during flow testing not with a propellant component, but, most frequently, with water. Therefore, by examining the results obtained during flow tests, it is necessary without fail to consider the density of the propellant and to insert the appropriate corrections.

If during the flow test with water, at nominal volumetric water flow rate Q_B l/sec a value of the hydraulic resistance of the valve is obtained equal to Δp_B, then during operation with the propellant with density γ_T at the same volumetric flow rate Q_B the actual value of the resistance in a self-modeling mode[1], will, as is known, be:

$$\Delta p_T = \Delta p_B \frac{\gamma_T}{\gamma_B} \approx \Delta p_B \gamma_T$$

(the density of the water is taken as $\gamma_B = 1$).

Let us find the mass propellant flow rate G_T, ensuring the given pressure differential Δp[2].

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[1]A self-modeling mode begins with rather large Reynolds numbers for the liquid flow; as a rule, during flow testing of valves it can always be assumed that an auto-modeling mode exists.

[2]Here we are speaking of the difference in static pressures before and after the valve.

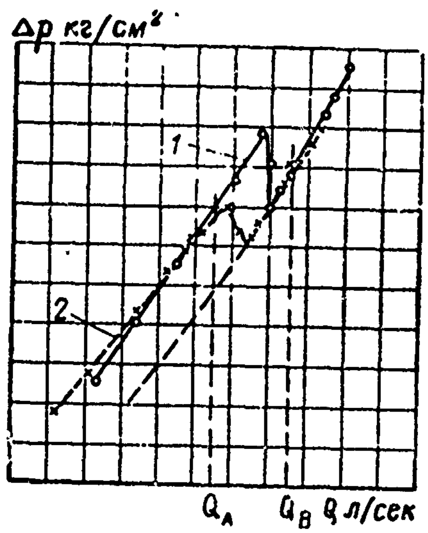

Fig. 6.28. Valve characteristics Δp = f(Q), obtained when a seal ring is reversed: 1 - flow test with an increase in the flow rate; 2 - flow test with a decrease in the flow rate.

Since

$$\Delta p = \xi \frac{\gamma v^2}{2g},$$

and v = Q/F, where F is the flow passage cross-sectional area, then

$$Q = F\sqrt{\frac{2g}{\xi}\,\frac{\Delta p}{\gamma}},$$

or, having designated $F\sqrt{\frac{2g}{\xi}} = k_o$ we obtain

$$Q = k_0\sqrt{\frac{\Delta p}{\gamma}}.$$

For water

$$Q_в = k_0\sqrt{\frac{\Delta p_в}{\gamma_в}}.$$

The coefficients of hydraulic resistance ξ for water and for propellants during automodel mode coincide, therefore the coefficient k_0 does not depend on the kind of liquid; then

$$Q_{\text{T}} = k_0 \sqrt{\frac{\Delta p_{\text{T}}}{\gamma_{\text{T}}}}.$$

The equivalent water flow rate, ensuring an equal-drop flow-test mode, i.e., a mode in which $\Delta p_{\text{B}} = \Delta p_{\text{T}}$, is determined from the relationship

$$\frac{Q_{\text{T}}}{Q_{\text{B}}} = \frac{k_0 \sqrt{\Delta p_{\text{T}}/\gamma_{\text{T}}}}{k_0 \sqrt{\Delta p_{\text{B}}/\gamma_{\text{B}}}} = \sqrt{\frac{\gamma_{\text{B}}}{\gamma_{\text{T}}}}.$$

Hence

$$Q_{\text{B}} = Q_{\text{T}} \sqrt{\frac{\gamma_{\text{T}}}{\gamma_{\text{B}}}}; \quad Q_{\text{B}} = G_{\text{T}} \frac{1}{\sqrt{\gamma_{\text{T}}\gamma_{\text{B}}}}.$$

Assuming $\gamma_{\text{B}} \approx 1$, we get

$$Q_{\text{B}} = Q_{\text{T}} \sqrt{\gamma_{\text{T}}};$$

$$Q_{\text{B}} = G_{\text{T}} \sqrt{\frac{1}{\gamma_{\text{T}}}}; \qquad G_{\text{T}} = Q_{\text{B}} \sqrt{\gamma_{\text{T}}}.$$

Thus, if for a valve we have the graph $\Delta p = f(Q_{\text{B}})$, obtained by means of a flow test of the assembly in water, then in order to obtain the dependence $\Delta p = f(G_{\text{T}})$ operating on a propellant component, it is sufficient to replot the axis Q, after multiplying the abscissa of each point by the value $\sqrt{\gamma_{\text{T}}}$. In other words, it is necessary to scale μ along the X axis:

$$\mu = \sqrt{\gamma_{\text{T}}}.$$

The component flow rate, at which valve opening begins, is similarly calculated.

The moment of the start of valve opening for operation on main-stage mode is in this case determined:

a) by the pressure p_2 behind the valve;

b) by the drop in the static pressures on the disk Δp,

c) by the velocity pressure head $\Delta p_Д$, acting on the disk (in practice, during flow tests $\Delta p_Д$ is usually not measured because of its small quantity).

The pressure p_2 behind the valve is fixed on the basis of analysis of the valve's operation, and its value is not connected with the liquid density. This value is not subject to change when converting from water to the propellant component.

The pressure differential Δp is determined by the flow rate and by the density of the liquid and, as is shown above, for an equal-drop mode

$$Q_т = Q_в \sqrt{\frac{1}{\gamma_т}}.$$

Thus, if we put to the side for now the problem of the dynamic head, then we can state that if during the flow tests with water the valve began to open at point A with counterpressure p_2 and water flow rate $Q_в^A$, then the beginning of opening of the valve when handling a propellant component with the same counterpressure p_2 will take place with sum flow rate $Q_т^*$, where

$$Q_т^* = \frac{Q_в^A}{\sqrt{\gamma_т}}.$$

The value of $\Delta p_{\text{д}}$ is expressed thus:

$$\Delta p_{\text{д}}=\frac{\gamma(\Delta v)^2}{2g},$$

where Δv is the difference in velocities of the liquid before and behind the disk.

When handling water

$$\Delta p_{\text{д}_{\text{в}}}=\frac{\gamma_{\text{в}}(\Delta v_{\text{в}})^2}{2g},$$

and when handling a propellant component

$$\Delta p_{\text{д}_{\text{т}}}=\frac{\gamma_{\text{т}}(\Delta v_{\text{т}})^2}{2g}.$$

But what will the dynamic head equal when operating with a propellant component at volume flow rate $Q^*_{\text{т}}$?

Since

$$Q^*_{\text{т}}=Q^A_{\text{в}}\sqrt{\frac{1}{\gamma_{\text{т}}}},$$

then

$$v^*_{\text{т}}=v^A_{\text{в}}\sqrt{\frac{1}{\gamma_{\text{т}}}} \text{ и } \Delta v^*=\Delta v^A_{\text{в}}\sqrt{\frac{1}{\gamma_{\text{т}}}}.$$

Thus,

$$\gamma_{\text{т}}(\Delta v^*_{\text{т}})^2=(\Delta v^A_{\text{в}})^2;$$

hence

$$\Delta p^*_{\text{д}_{\text{т}}}=\Delta p_{\text{д}_{\text{в}}},$$

i.e., the forces from the dynamic head for water at flow rate Q_B^A and for a propellant at flow rate $Q_T^* = Q_B^A \sqrt{\frac{1}{\gamma_T}}$ will be precisely equal. Consequently, the dynamic head with the conversions of the results of the flow tests using water for the propellant cannot be taken into consideration simply.

Since the value of opening of the valve h at a given counter-pressure p_2 and flow rate Q, as was shown above, is determined by the pressure drop on the valve Δp, then everything said above about conversions for the propellant component can be extended, naturally, also to the graphs of h = f(Q).

Thus, the general conclusion can be drawn that in converting water-to-fuel flow test data, in order to obtain the dependences of $\Delta p = f(G_T)$ and $h + f(G_T)$, it is sufficient only in the experimental graphs of $\Delta p = f(Q_B)$ and $h = f(Q_B)$ to change the numbers on the flow rate axis on the scale $\mu = \sqrt{\gamma_T}$, or, more precisely, on the scale $\mu = \sqrt{\frac{\gamma_T}{\gamma_B}}$., where γ_B is the water density at the test temperature, while γ_T is the propellant density at the given temperature.

It is much more difficult to determine the equivalent water flow rate for simulation of the value of the hydraulic impact, produced during operation of the valve, installed in the engine.

In the first place (and this is the main thing), it is difficult, and in practice almost impossible, to simulate the assembly's working conditions - the length and rigidity of the tubing from the source of the hydraulic impact to the free mirror of the liquid.

In the second place, it is unclear what is meant by equivalent flow rate? If we proceed from Zhukovskiy's formula, $\Delta p_\Gamma = \rho va$ then to obtain the equality of the pressure increase $\Delta p_{\Gamma_T} = \Delta p_{\Gamma_B}$, the following condition must be ensured:

$$\varrho_{\text{т}} v_{\text{т}} a_{\text{т}} = \varrho_{\text{в}} v_{\text{в}} a_{\text{в}},$$

where the subscripts "в" refer to water, while the subscripts "т" refer to propellant; then

$$v_{\text{в}} = v_{\text{т}} \frac{\varrho_{\text{т}} a_{\text{т}}}{\varrho_{\text{в}} a_{\text{в}}}, \quad \text{or} \quad Q_{\text{в}} = Q_{\text{т}} \frac{\gamma_{\text{т}} a_{\text{т}}}{\gamma_{\text{в}} a_{\text{в}}}.$$

However, here the forces acting on the moving system of the valve when handling water will be different than when handling propellant, since these forces, as a rule, are determined, if not completely, then partially, by the pressure differential Δp, acting on the moving system, and the value of the drop is recalculated by the ratio $\sqrt{\gamma_{\text{т}}/\gamma_{\text{в}}}$. A change in the value of the forces acting on the moving system causes a change in the time of opening of the valve, as a result of which during the transition from water to propellant in place of complete hydraulic impact we may get incomplete impact, or vice versa (depending on the value of $\gamma_{\text{т}}$).

Thus, the value of the hydraulic impact is connected with the pressure differential of the valve. It is impossible to simulate simultaneously the value $\Delta p_{\text{г}}$ and the value Δp (equivalent to the opening time), since it is impossible to find the similarity criterion, which would satisfy both requirements simultaneously.

Therefore, in water flow tests one should reproduce the valve's riggering time as the value which determines the character of the transient processes, a much more reliable value and one much easier to simulate. Hence it follows that the equivalent water flow rate in the determination of the valve's opening time should mean

$$Q_B = G_T \sqrt{\frac{1}{\gamma_T}}.$$

The hydraulic impact value for a valve handling propellant should be estimated, proceeding from the value of the hydraulic impact, obtained with water. If we succeed in completely simulating the lengths and rigidities of the tubing when handling water and when handling a propellant component, then the hydraulic impact value in propellant Δp_{Γ_T}, proceeding from the measured value of the hydraulic impact in water Δp_{Γ_B}, can be calculated thus:

$$\Delta p_{\Gamma_T} = \Delta p_{\Gamma_B} \frac{\gamma_T Q_T a_T}{\gamma_B Q_B a_B} \approx \Delta p_{\Gamma_B} \frac{G_T a_T}{Q_B a_B}.$$

Here one should take into consideration the dependence of the pressure propagation rate in water a_B and in the propellant a_T on the temperature, as well as the effect of the pressure on the value of a_B in tap[1] water.

In general, in flow tests conducted in water, depending on the purpose of the investigations, the term "equivalent water flow rate" can be understood to mean this or some other value.

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[1]As a result of dissolved gases in it.

CHAPTER 7

THE PROCEDURE FOR CONDUCTING TESTS OF PROPELLANT VALVES

7.1. PROCEDURE FOR PERFECTING THE VALVE DESIGN

Even with a carefully worked out assembly design special finishing of it is essential, in order that it completely respond to the imposed requirements. Therefore, in the creation of a new valve or with a major form change of the existing one it must undergo a number of comprehensive checks - both autonomous checks, and checks during operation on the engine. The totality of these tests is called the "finishing" of the assembly, or finishing tests.

The initial perfection of the assembly is carried out in autonomous tests. In these finishing tests a study is made of the peculiarities of operation of the valve in all possible (with maximum degree of approximation) operating conditions. Design or manufacturing technology defects are exposed and eliminated.

However, the autonomous valve test, independent from the engine, is still not sufficient to judge completely the valve's working efficiency. The specific engine operating conditions - vibrational and impact loads, high pressures and temperatures,

the influence of chemical activity of the propellant components, high flow rates, pressure pulsations and so forth - in their entire totality, naturally, cannot be completely simulated on the test benches of an independent test of the assemblies. But this does not mean, as a rule, that such tests are useless. On the contrary, a number of autonomous tests of individual perimeters of the valve and the totality of checks of the working efficiency of the assemblies during simulation of only one or several factors of the possible combinations of working conditions permit, right in the early stage of finishing, the elucidation of mistakes which have crept in during the designing or manufacturing of the assembly.

The cost of a single test of a modern engine of high thrust is very great. This includes the cost of the propellant, the cost of amortization of the equipment, the cost of developing and analyzing the results of operation and so forth, and, finally, the cost of the equipment itself. The unsuccessful outcome of an engine test, caused by an inadequately finished propellant valve, results in not only the demise of the machine, but may also lead to the partial destruction of the even more expensive test bench, the reconstruction of which will entail, besides the financial losses, a great expenditure of time. Therefore, even while conducting the first adjustment tests of the engines it is necessary to have a certain degree of confidence in the working efficiency of all the assemblies of the engine, including such critical assemblies as propellant valves. And this confidence can be based only on an adequate number and an in-depth analysis of the conducted autonomous tests.

Moreover, one must not forget that the creation of the test bench for autonomous tests of the assemblies on which the test conditions completely (or even only partially) correspond to actual operating conditions, is a rather difficult thing. A

change in the parameters of newly developed engines leads to the rapid obsolescence of the test benches; one must be careful that the test bench not become useless before it is finished, in connection with the fact that the assemblies for which it was designed are already finished, while the parameters of the assemblies of a new engine have been boosted to such a degree, that they cannot be achieved on the test bench on hand. Therefore the parameters of the newly developed test benches must be given the most careful attention, and the prospectives of the forthcoming operations must be evaluated in depth.

With respect to the degree of finish of the construction and the production preparedness, autonomous finishing tests are grouped into the following stages:

a) investigative and experimental-finishing works;
b) structural-finishing or preliminary finishing tests (PFT);
c) final finishing tests (FFT).

The investigative and experimental works have the purpose of clarifying some specific peculiarities of operation of the assembly, of clarifying individual particular problems. As examples of experimental tests we can point to: the determination of the effect of manufacturing accuracy of parts (within the tolerance limits) on the time of opening of a pneumatic valve under various temperature conditions; the determination of the wedging forces of the valves in pyrotechnic assemblies with different explosive cartridges and with different strength of the parts; the determination of the effect of the form of coating on the service life of a pneumatic valve.

In this, the first stage of finishing it is advisable to make special very simple simulators of the assembly's subunits for clarification of individual questions. Thus, it is advantageous

to make a simulator of the flanged joint of the valve[1], in which one can check the working efficiency of the seal, perfect the degree of tightness of the connection, and select the sealing material, compression value and so forth.

These simulators should be tested at various temperatures (the limits of which sometimes exceed the range of operating temperatures), after buffeting, during vibration and so forth. The minimum and maximum allowable tightening, determined during the checks of the simulators, should then be checked in the PFT, having specified assemblage of the assemblies with the extreme values of tightening. In the simulators one can determine the service life of the moving seals, the service life of the bellows, the required amount of lubrication and so forth. However, due to the complexity of manufacture of such simulators, these problems are frequently checked directly on the valves.

Examples of investigative works are: the determination of the coefficients of hydraulic resistance of the valve's flow passage; the study of the effect of various factors on the time of triggering of the valves; the selection of the profile and type of seat to ensure the longest service life of the rubber seal with a certain degree of hermeticity, etc.

Research work in the initial period of creating the assembly aids in the proper determination of the valve's layout, selection of the dimensions of the flow passage cross sectional area, materials of the parts, etc.

Experimental tests can be carried out at any stage of the finishing of a valve and engine - in order to clarify a detected peculiarity of its operation and to solve any problem which might

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[1]This is done with large diameters of the flanged joint and high pressures of the sealed propellant. Sometimes simulators of joints of conduits which operate under severe conditions are made.

have arisen in the finishing process. However, the appearance of such problems at the end of finishing or in the course of series operation testifies, as a rule, to the insufficiently high quality of the finishing process for the assembly, or to some kind of oversights in the programs of the design-finishing operations.

A whole complex of problems connected with the operation of the valves must be clarified during the preliminary (design-finishing) tests. This is the main stage of finishing. The PFT program provides for the study of the tr nsient processes of the opening and closing of pneumatic valves during all possible operating conditions, with minimum and maximum temperature values, with extreme pressure and flow rate values, and with extreme stress values (for the electrically operated valves).

The program of preliminary finishing tests of propellant valves must include:

- flow testing in order to determine the hydraulic resistance and its stability;
- the determination of the pressure value in the controlling cavity of the pneumatic valve at the moment of the initiation and cessation of movement of the locking mechanism for closing and opening (with given pressures in the liquid cavity);
- the determination of the pressure values in the liquid cavity (with a given pressure in the controlling cavity) at the moment of initiation and cessation of movement of the locking mechanisms;
- the determination of the minimum required stress for normal triggering of the electrically controlled valves (with a given propellant pressure);
- the determination of the time of opening and closing of the valve (the time from the moment of the command to trigger the EPV or explosive cartridge until the beginning or until the end of movement of the moving system, or of the beginning and end of the rise in pressure on the valve outlet).

These times depend in many respects on the test conditions and the test-bench system. It is therefore important that all these conditions, even if not completely simulating the valve's operation on the engine (this is frequently impossible), at least be strictly identical during all the tests of the various modifications of the assembly, independent of temperature, date and site of the test, etc.;

- determination of the hermeticity of the seal in all moving[1] and stationary connections (by checks with compressed air, an air-helium mixture and with actual propellant);

- the determination of the resistance of the assembly's parts (including the commercial rubber ones) under various conditions of the environment and when handling propellant;

- a check of the triggering of the explosive valve with increased and decreased charges of the explosive cartridge as well as with extreme values of the thickness of the shorn-off part;

- determination of the magnitude of hydraulic impacts during triggering of the assemblies;

- determination of the possible service life for triggerings of repeat-action valves during operation with compressed air and with propellant components, with the maintenance of their complete working efficiency;

- vibration tests (transportation);

- check of the working efficiency of the assembly after vibration tests;

- check of the working efficiency of the assembly with a constantly acting overload (simulation of the forces of acceleration of a flight vehicle);

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[1]It is desirable to check the hermeticity and working efficiency of such assemblies, the moving and nonmoving seals of which have extreme limits with respect to the interferences of the rings, seal rings and other sealing parts.

- check of the working efficiency of the assembly with overloads as a result of vibrations;
- determination of permissible deformations and stresses in the parts;
- determination of the assembly's margin of safety;
- determination of the quality of the welded seams;
- check of the influence of external vacuum;
- check of the effect of moisture from the environment;
- check of the ohmic resistance and strength of the insulation of the electromagnets of the electrically operated valves;
- determination of the force developed by the electromagnets.

It is obvious that an actual program of finishing tests includes only those checks of those enumerated above, which correspond to the operating and construction conditions of this or that specific valve. In individual cases it may be deemed necessary to conduct even other types of checks or tests, not mentioned above.

In the PFT checks are made not only of the working efficiency of the assembly within the limits of the operating conditions, cited in the technical specifications for the engine, but also within broader limits - attempts are made to determine the limits of possibilities of the assembly with respect to service life, pressures, flow rate, temperature, strength and so forth.

In certain cases (most frequently for the finishing of explosive valves) assemblies, the most critical parts of which are made with limiting divergences from the nominal dimensions, are manufactured with this purpose in mind.

For example, it is possible to make a shorn collar T (see Fig. 2.15) of maximum thickness and from the strongest metal (but conforming to All-Union State Standard GOST or to the Technical Specifications [TU]) and to check the operation of the wedge with

the weakest explosive cartridge. Since it is difficult to produce such a metal quickly, one can manufacture the shorn collar from existing material and, taking into consideration its actual strength factors, one can increase the width of the collar, in such a way that it will be of equal strength to the collar made from the strongest material. Similarly, the weakest collar is checked (with a strong explosive cartridge). However, this version should first be tested for vibration strength - to see whether the partial shearing of the bead will occur even before the triggering of the explosive cartridge.

When conducting the PFT one should not only note the conformance of the parameters to the imposed requirements[1], but also scrupulously fix the true values of all the parameters and trace the change in parameters with the change in test conditions[2]. There should be widespread use of objective means of measurement - recording of the processes on a loop oscillograph and autographic apparatus, photographing of the instrument readouts, and filming.

However, it is important not only to ascertain a defect; it is much more important to precisely determine the location of the defect and its cause. Sometimes this is far from simple to do.

For example, let us assume that during the checking of the total hermeticity of bellows 4 (see Fig. 2.8) and gasket 14 when pressure is supplied to the controlling cavity a leak is discovered; where it is - in the gasket or in the bellows - is impossible to determine immediately. It is necessary in this case to make a small special operation: it is first necessary to check the

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[1]The requirements are refined from the results of the PFT; the requirements may be different in different stages of the checks.

[2]This helps in analyzing the reasons for a defect.

leak when pressure is supplied to the liquid cavity; if the leak remains unchanged, then one can assume that the gasket is nonhermetic (if the bellows is nonhermetic, the leak will have a different value, depending on whether the pressure is supplied inside or outside the bellows - due to the different deformation of the corrugations of the bellows). To be convinced of non-hermeticity of the gasket, one must tighten the connection. If the leak disappears, then it means that the leak occurred through the gasket (however, the bellows should nevertheless be tested individually at the end of the tests, in a special device).

Let us examine another example. Let us assume that the pressure differential Δp in the preliminary stage during the checks of one or another valve (see Fig. 2.10) with a constant flow rate varies from test to test. If analysis has shown that the measurement is made correctly, with sufficient precision, then the reason for the defect may either lie in the jamming of parts of the valve, or in an excessively soft seal in this disk sample, as a result of which the indentation of the seat on the seal varies. To check the assumption of softness of the seals one can compare the degree of countersinking of the seat into the rubber with various controlling pressures on this part and on several others. If the countersinking in the defective assembly significantly exceeds the countersinking value in the other assemblies, then the expressed assumption becomes probable. An example of an operation to determine the causes of a defect may be the case with the reversal of seal rings 1 (see Fig. 6.27), described in section 6.3.

Sometimes there may be a set of assumed reasons for a defect. For example, the opening of one sample of a propellant valve (see Fig. 2.3) occurs more slowly than the opening of other samples of this same valve. What may be the reason? Increased friction of the seal rings? An improperly manufactured spring? Jamming of the moving system as a result of poor centering of the parts?

Improper manufacture of the flow passage cross sectional areas in a relief valve?

The procedure for clarifying the reasons for a defect will in each individual case be different. However, in all cases one must be convinced that the test procedure, measurement accuracy and the system of test-bench equipment cannot introduce errors which distort the check results. If, to determine a defect, it is essential to disassemble or to change the form of the test conditions, then it is necessary in each subsequent check to change only one factor: either only the part, or only the test conditions. When changing several factors at once (when both replacing parts and changing the test conditions) the reason for a defect, in case of its disappearance, cannot be clarified. The spontaneous elimination of a defect not understood by the tester is the most unpleasant occurrence during finishing, since this means that the defect can reappear at any moment. Thus the simultaneous change of several factors is not permissible.

In an example with retarded valve opening the sequence of operations for clarifying the reasons for a defect may be assumed to be as follows (in each stage the opening time value is checked):

a) replacement of only the relief valve unit;

b) tests at low temperature (two or three additional assemblies, which conform to the TU, are simultaneously checked under identical conditions);

c) addition lubrication of the seal rings;

d) disassembly of the unit without the replacement of parts;

e) disassembly with the replacement of the seal rings;

f) disassembly with the replacement of the spring (the characteristics of the old and new springs are investigated);

g) disassembly with the replacement of the rod (the diameters of the rods are measured) and so forth.

If there are substantiated assumptions concerning the reasons for defects, then the assumed reasons are studied first of all. For example, if the technology for the lubrication of the seal rings is not perfected, then a check with additional lubrication of the seal rings is carried out first of all.

A very important step in the PFT is the dismantling of the assemblies. It is necessary to know how to detect (and to carefully locate) all the defects or changes, which have occurred with parts, for which a certain experiment is required. Then the parts must be measured to determine important clearances (sometimes to detect wear).

Defects detected in the parts must be correctly explained, and the reason for their occurrence must be understood. Let us assume that the rubber on an end seal is strongly scored, that the impression is too sharp, and that the rubber is on the verge of destruction. The reason for the poor state of the seal may be:

a) low quality of the rubber (only in the given part);

b) improper (with skewness) assemblage, as a result of which the seal is subjected to additional loading;

c) improper manufacture of the seat profile;

d) excessive (exceeding that allowable by the TU) cooling or heating of the assembly during the process of testing;

e) deviation from the established regime during vulcanization of the rubber.

These are accidental reasons. However, design errors leading to the same defect may also take place:

a) improperly selected seat profile, as a result of which high specific loads act on the seal;

b) improperly selected brand of rubber;

c) incorrectly designated depth or width of the channel beneath the rubber seal;

d) too great an operating service life designated;

e) improper manufacturing technology of the parts (as a result of which overheating of the rubber takes place during finishing).

An analysis of the test results of other samples of the valve (especially of those manufactured during the same time period) and a comparison with the test results of analogous designs aid in the solution of the problem of specific reasons for the discovered defect.

The quantity of samples of assemblies necessary for conducting the PFT is difficult to state in general - this problem is solved in each individual case. One must clearly see that the successful conducting of tests of ten or twenty valves cannot justify the failure of a single valve, manufactured from the same sketches, during operation, if a deviation from the technical specifications and records was not detected with complete accuracy.

The presence of a failure in operation of only one of ten tested assemblies reduces the theoretical reliability of the assembly so much, that it excludes the possibility of taking the results of such tests as satisfactory. It is in this case necessary to analyze the defect, discover its reasons, and take measures which unconditionally eliminate the repetition of the defect.

The more complete the program of preliminary finishing tests, the lower is the probability of the occurrence of defects in the assembly's design during its operation, the lower the possibility of the emergence of experimental work in the final stage of finishing involving loss of time, and sometimes also the necessity of changing the valve's construction.

On the basis of analysis of the results of the PFT measures must be taken - structural, technological, and sometimes also organizational - for the elimination of any defects uncovered.

To check the effectiveness of the measures taken repeated PFT should be conducted, however in certain instances small-scale tests are adequate.

. Final finishing tests (FFT) are conducted after the successful completion of the entire program of the PFT.

The assemblage of the valves for the FFT is carried out in strict conformance with published technical specifications and records; all the dimensions of the parts must also conform to the blueprints and other official documents. All the peculiarities of assemblage for the FFT must be carefully reflected in the accompanying technical documentation. The manufacture of parts and the assemblage of valves are not allowed, if based only on considerations of the advisability of this or another deviation from the technical specifications and records (even if they are unconditionally correct); it is possible then that these correct solutions will for some reason or other not be realized in the blueprints, and the assemblies, put together for the FFT, will differ from the following, commercial samples. Then the successful completion of the FFT will not testify to the reliability of the commercial assemblies.

The FFT program ordinarily is not as comprehensive as the PFT program - it includes only a strict check of the valve for conformance to technical specifications, without the determination of the limiting possibilities of the assemblies. Therefore, the service life of the pneumatic assemblies in the FFT is limited only to the value given in the TU; the parts are manufactured within the tolerance limits, given by the blueprints, without striving to check their extreme values.

Final finishing tests control not only the valve's construction, but also the degree of finishing of the manufacturing technology.

If improper manufacture of one part or another turns out to be the reason for an unsuccessful test, then, even though the construction of the assembly will not be discredited by this, the FFT are not considered, because the results of the tests mean that production is not ready for the manufacture of the assemblies in conformance with the imposed requirements. In principle, the nonpreparation of production should also be stated in the case where the failure of a part is detected only when measuring the parts after the successful completion of the test program; however, usually in such cases the reasons for the deviations from the blueprints are analyzed, along with the possibility of the repetition of the defects in the future, the effect of this deviation on the valve's operation, and from this analysis and from the results of the FFT a final solution is made. Sometimes this solution may take the form of the expansion of the manufacturing tolerance of the part.

To evaluate the results of the FFT the condition of the parts with the assemblies disassembled is important. Even in the case of complete conformance of the valves to all the imposed requirements with respect to hermeticity, triggering times, hydraulic characteristics and so forth, the presence of defects in the material can serve as the basis (generally for repeat-action valves) for a negative conclusion on the construction or manufacturing technology of the assembly. If when dismantling the valve after the wearing out of the given service life a poor condition of the rubber seal has developed - the removal of rubber, too deep an impression from the seat - then where is the guarantee that another sample of the valve will not lose its hermeticity during the very process of the service life? In any case it is essential to reveal the cause for the defect and to try to eliminate it. (By the way, it should have been detected in the process of the PFT).

The concluding stage of the FFT must be the check of the working capacity of several samples of valves on experimental engines, which have passed the adjustment tests.

Ordinarily after the completion of the FFT official word is issued on the finishing of the assembly, in which all the finishing operations are annotated; an analysis of the working capacity of the assembly is given and the peculiarities which arose during its finishing are cited. In this document all the experimentation for the finishing of the assembly must be summarized and the technical specifications, and the methods for control and acceptance of the valve must be refined. The sum total of the report of successful finishing tests must be the conclusion that the valve can be operated in an engine.

With the successful completion of the final finishing tests, in order to check (or to determine) the possible storage life of the assembly with the maintenance of its working capacity, several samples of the assemblies, pre-tested for their conformance to the given technical specifications, are laid away for prolonged storage under sealed or warehouse conditions. The storage conditions (temperature, humidity and so forth) and the check regulations are specified in special documents.

To more quickly determine the service period of efficient operation of valves, which have critical commercial rubber parts (seal rings, rings, valves with vulcanized rubber seals), artificial aging methods are employed (see Chapter 3), which permit us to obtain data on the problems which interest us in relatively compressed time periods. Very frequently a check of commercial rubber parts by the artificial aging method is carried out in parallel with the FFT of the valves.

7.2. CERTAIN RULES FOR CONDUCTING THE TESTS

When conducting the tests of valves it is necessary to maintain a number of conditions, the observation of which ensures reliability of the obtained results and the maintenance of the working efficiency of the assemblies after the tests. Enumerated below are the simplest fundamental conditions for conducting ordinary tests of propellant valves.

7.2.1. Assurance of Cleanliness of the Test-Bench Connection Lines

A high degree of cleanliness of all the test facility systems - conduits, reservoirs, measuring devices and test bench assemblies - is essential, so as to guarantee the hermeticity of the locking mechanisms of the valves of any design and of the rubber seals (seal rings and other rings), to avoid wedging of the parts of the moving system of valves, operating with small diametric clearances, and to guarantee explosion safety when testing with propellant components or during the subsequent operation of the assemblies on the engine.

Before the assemblage of a newly created test bench it should be thoroughly cleaned of scale; reservoirs and large-diameter conduits - by means of sandblasting; straight small-diameter conduits - by pumping corundum powder[1] through them (for bent tubing this is dangerous in view of the possible thinning of the walls). Widespread use is made of chemical treatment of parts and conduits - pickling and passivation. After mechanical cleaning and in the subsequent preventive operations degreasing

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[1] Excluding tubing designed for hydrogen peroxide.

of the reservoirs, conduits, and parts of the test bench facilities is carried out by washing or wiping them down with benzine, carbon tetrachloride, freon-13 or dichloroethane. Flanges, silencers, adapters, manometers, sensors and so forth are also subjected to degreasing. Sometimes ultrasonic cleaning of parts is used.

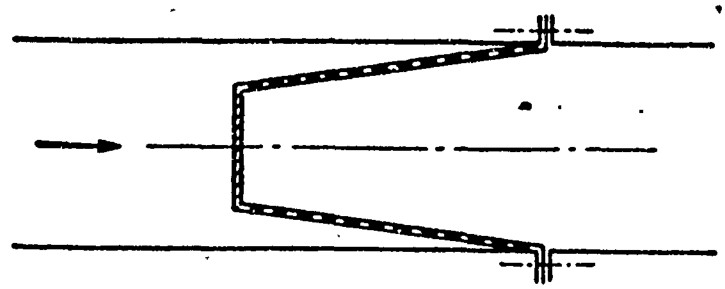

Fig. 7.1. Diagram of a filtering device in a conduit.

To prevent foreign particles from getting into the assembly, the test facility uses filters - either directly in front of the entry into the tested valve, or in the main conduit lines in front of the test bench facility.

For the filtration of water and a number of propellant components small-mesh and very small-mesh metal grids with serge and smooth networks (according to GOST 3187-65) No. 24 and No. 64 are used. In certain cases even finer meshes are employed - up to number 160. The grids are manufactured from stainless steel, glass or nickel wire of appropriate diameter. The water flow rate through a square decimeter of the filtering element must not exceed 1-3 l/sec. In order to achieve a low water flow rate through the filter (not exceeding 3-6 m/sec) with a high flow rate in the tubing, filters are set up according to the diagram shown in Fig. 7.1. The pressure drop on the filter is not allowed to exceed 2.5 at.

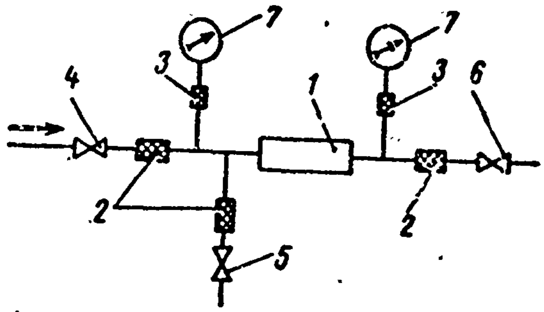

Fig. 7.2. Arrangement of filters to ensure a high degree of purity of the tested assembly: 1 - tested assembly; 2 - filters; 3 - filters for the manometers; 4, 5, 6 - valves; 7 - manometers.

To purify compressed air or nitrogen chamois or ceramet filters are installed, and sometimes also filters manufactured from the above-mentioned smooth-mesh grids; sometimes filters made from capron fabrics are used. The hydraulic resistance of the filters for air at nominal flow rate ordinarily does not exceed 2-3 at.

In the tests of critical assemblies handling compressed gas, in the case where there are high requirements for purity, special filters are set up on the lines leading to the manometers, sensors, and also at the outlet from the assembly with the aim of protecting the assembly from contamination during the discharge of air pressure through valve 5 (Fig. 7.2).

7.2.2. Elimination of the Possibility of Precipitation from Liquids or Gases

Besides careful filtering of the working medium at the inlet to the equipment or valve, for some assemblies with valve mechanisms it is essential to eliminate the possibility of the precipitation of solid particles, deposited on the working surfaces of the tested assemblies, from liquids or gases during the tests.

Thus, for flow tests of assemblies with small clearances, instead of common drinking water, which contains calcium and magnesium salts which form precipitates, permutite (Na-cationate) water, in which these salts are easily replaced by sodium salt

solutions[1], are employed. In order to reduce the content of oil impurities in compressed air, gaseous nitrogen, obtained by the vaporization of liquid nitrogen, is sometimes used in place of compressed air.

7.2.3. Prevention of Corrosion of Parts During Tests

During flow tests of assemblies with water circulation in a closed system (see Fig. 6.8) it is advisable to add potassium dichromate to the water - at a rate of about 3 grams of potassium dichromate to 1 liter of water. The presence of the potassium dichromate practically excludes corrosion of aluminum, bronze and steel parts with the total filling of the assembly's internal cavities with water (corrosion may appear on the border of the water divider, on the water line). However, potassium dichromate cannot be used when testing valves designed to handle hydrogen peroxide.

After flow testing the assemblies with water, they should be completely dried as quickly as possible. An incompletely dried valve cannot be left without water after flow testing - in this case corrosion occurs significantly more intensively, than when the valve is placed in water. In many cases a maximum permissible time between the ends of the flow test and the start of drying is specified.

It is desirable to dry the valves in a vacuum (with a residual pressure of 5-10 mm Hg.) at a temperature of 50-60°C, with a periodic (2-3 times) pressure increase up to atmospheric.

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[1]Permutite water causes the corrosion of parts manufactured from aluminum and its alloys. Therefore, the conducting of tests in permutite water requires a high degree of work organization, careful and undelayed drying of the assembly after the work has been completed.

A higher temperature in the case where commercial rubber parts are used is not desirable. The drying time is indicated in the technology. With a complex cavity configuration, where there are zones which can trap moisture, when using materials especially noncorrosion-resistant to water, the entire internal cavity of the valve should be washed down with alcohol or acetone, dissolving the moisture and quickly evaporating, before it is dried.

When testing assemblies which handle propellant components, much attention should be paid to the selection of the material of the parts, used in the test-bench valves, the material of the branches, adapters, coupling fittings and so forth. The selection of the material for these parts must be even more rigorous with respect to their corrosion resistance, then for parts of assemblies of the engine, since they come in contact with the product for an immeasurably greater time.

With prolonged contact with oxidizers based on nitric acid aluminium alloys, employed in the housings of valves, filters and so forth, are subject to corrosion. The products of this corrosion are carried off with the propellant into the tested valve. The process of corrosion proceeds with special intensity with an increase in the moisture content. An increase of moisture in the oxidizer above the permissible limit sharply accelerates the corrosion process, and therefore it is necessary to periodically take a chemical analysis of the product, and to watch its composition.

7.2.4. The Elimination of Pressure Measurement Errors

In all cases measurement error must not exceed 30% of the tolerance for the value of the perimeter. To ensure high accuracy, the value of the measured pressure is sometimes located within the limits of the last third of the manometer scale (although this contradicts the general recommendations).

When checking the quality of the transient processes with compressed air using manometers (sensors) the length and the hydraulic resistance of the connected conduits are very important. During all the flow tests with water and components for during gas blasting it is important to precisely observe the recommendations of "rules 28-64 for the measurement of the flow rate of liquids, gases and vapors using standard diaphragms and nozzles"[1]. Incorrect organization of the pressure measurement - the welding of the measuring connector tube not perpendicularly to the axis of the tubing, its placement close to the site of abrupt expansion of the cross section, on a curvilinear or conical section of the tube, nonhermeticity of the measurement tubing and so forth - can significantly distort the results of the tests. It is desirable to run the pressure takeoffs through annular chambers.

With a pulsating character of the pressure change, when the pressure is measured with a manometer, it is necessary to put dampers in front of the manometer - according to the diagram shown in Fig. 7.3, in which either gas or liquid can be fed to tube a; tube b must in all cases be filled with liquid.

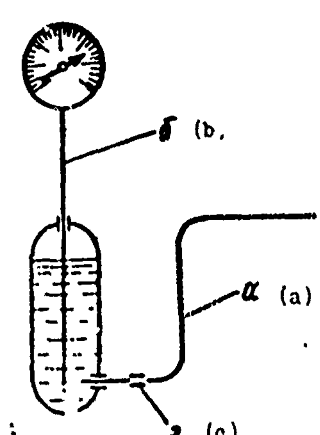

Fig. 7.3. Placement of a damper before the manometer: a, b - tubes; c - discharge nozzle

[1]Publishing House of the Committee of Standards, 1964.

Instead of discharge nozzle d it is desirable to install a capillary tube ensuring a laminar flow of the liquid through the capillary during "surges" of the flow. This avoids possible errors in determining the average pressure value during a pulsation[1].

The servicing of differential manometers with quick transition of the assembly to working mode and at h[illegible]solute pressures requires operating skill of the service personnel to avoid damage to the instruments or the discharge of mercury (in mercury differential manometers). The recommended layout for a differential manometer is shown in Fig. 7.4. The differential manometer is switched to steady-state conditions. During the transition to a mode valves 1 and 2 are open, while valves 3 and 4 are closed. After transition to a mode valve 3 opens, then valves 1 and 2 close, after which valve 4 is opened. Valves 1 and 2 duplicate one another - for a reliable hermetization of the line (sometimes a drainage valve is installed between them).

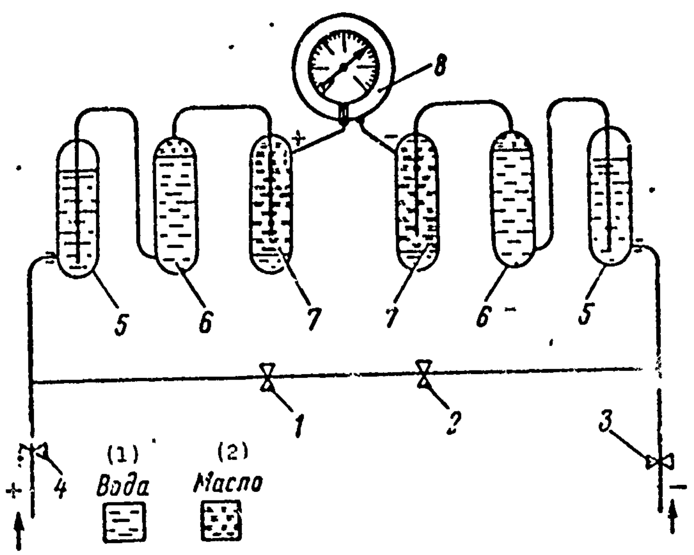

Fig. 7.4. Diagram of the installation of a differential manometer: 1, 2, 3, 4 - valves; 5 - dampers; 6 - small water cylinders; 7 - small oil cylinders; 8 - differential pointer manometer.
(1) Water; (2) Oil.

[1]This problem is studied in detail in work [17].

Bellows-type separators are installed in certain cases in front of the manometers when handling aggressive components. When working with differential manometers, and when the rigidity of the bellows separators introduce excessive errors as a result of the tests, no separators are used, and the small cylinders are filled with a neutral liquid which does not react with the given propellant. When handling liquid oxygen gasification of the oxygen is employed in the manometer tubes, preventing excessive cooling of the instrument; sometimes these tubes pass through a special heat exchanger.

7.2.5. Eliminating Errors when Determining the Nonhermeticity of the Exchanger.

In determining the nonhermeticity of a seal by means of removing air through a tube into water (see Fig. 6.7) it is essential to ensure strict invariability of the temperature during the process of measurement, especially with considerable volumes of the cavity, through which the leak is supposed to be.

An increase in temperature during the check of the hermeticity results in the fact that a (false) leak will be fixed, due to the expansion in air A. A temperature decrease can lead to the fact that nonhermeticity will not be detected. Therefore, if the temperature of the assembly differs from the temperature of the environment, it is essential to regulate the time of the check and to take measures to completely eliminate temperature fluctuations while determining the nonhermeticity. This is especially important in deep cooling. To control the temperature a thermocouple should be calked into the body of the valve and the temperature should be watched.

During temperature tests of large-scale valves it is necessary to ensure a constant temperature seal around the assembly, so that the temperature at various points on the valve will be identical. It is especially important to observe this condition when testing the assembly on components, in cases where the test temperature is close to the freezing or boiling temperature of the component.

When checking the hermeticity of stationary joints by the "aquarium" method, before putting the assembly into water or before filling with water the baths with the valve lowered therein, the assembly should be supplied with a small quantity of air pressure - about 0.2-0.35 at. The purpose of this check s to prevent the consequences of poor joining of the conduits, po r installation of the mufflers, as a result of which water might enter inside the valve.

7.2.6. Prevention of Cavitation During Flow Tests

The possibility of operation of the system under cavitation conditions should be eliminated. To avoid cavitation, it is sufficient to provide a counterpressure valve p_2 (pressure behind the assembly), which twice exceeds the value of hydraulic resistance Δp. However, sometimes high pressure p_2 is difficult to ensure, while cavitation may be absent even at very low p_2. In order to make sure there is no cavitation at a given flow rate, it is enough to change the counterpressure p_2; constancy of the value Δp (at the same flow rate) will testify to the absence of cavitation[1].

7.2.7. Consideration of the Actual Speed of Sound Propagation in Water

In investigating the value of hydraulic shocks which appear during valve triggerings in hydraulic facilities one should consider the actual value of the pressure propagation rate in water.

The speed of sound in water varies depending on the pressure as a result of the solution of gas in it. Besides the fact that

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[1]Naturally, with unchanged position of the moving system of the valve.

water itself contains various quantities of dissolved gases in it, during the process of tank pressurization the saturation of the water with air - airation of the water - occurs. The speed of sound in water reaches a maximum at a pressure of 12 at, and with a further increase in pressure remains constant (Fig. 7.5).

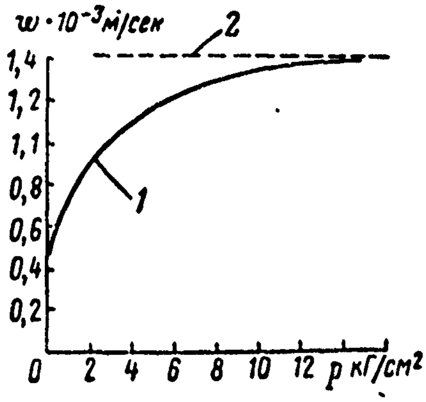

Fig. 7.5. The dependence of the sound propagation rate in water on the pressure.

In a test bench facility with pressure feed system for studying phenomena for hydraulic shock at low flow rates and pressures, but with high water speed, it is des'rable that the air act on the water through a rubber membrane.

7.2.8. Ensuring a given Sequence for the Passing of Electrical Instructions

When checking the operation of a system of explosive valves it is essential that the explosive cartridges trigger precisely at given moments in time. Sometimes it is important to ensure the triggering simultaneity of the explosive cartridges [or to check the effect on the operation of the system of the actual (insignificant) time difference in the explosion of the cartridges].

The required accuracy of the moment of supplying the signal is ensured by the appropriate timing mechanism, which acts on the explosive cartridges through the contacts of the electromagnetic relays.

In order to be able to check the continuity of the electrical circuits of each individual cartridge, each cartridge gets the command from a separate electromagnetic relay, or else from various contacts of the same relay. The effect of the operation of the relays also results in an actual time difference in the passage of the common (single) electrical instruction.

Before the tests one should carefully check the assigned timing sequence for the triggering of the explosive cartridges. For this, instead of the cartridges, simulators for them are connected to the plugs of the cable shaft and the sequence for the passage of instructions is recorded on an oscillogram (sometimes several times each). Only after receiving the assurance of reliable and stable operation of the system can the cable shaft be connected to the explosive cartridges.

The electrical system of a console permitting us to check the continuity of the circuits of two explosive cartridges (with a duplicated feed system for each cartridge), to record on an oscillograph a sequence for passing instructions and to detonate the cartridges is depicted in Fig. 7.6.

7.2.9. The Attachment of the Assemblies on the Vibration stands

In checking the assemblies for the effect of vibration on them (on vibration stands), and also in checking the effect of transportation (on jostling stands) there arises the problem of the method of fastening the valves to the table of the test bench, since the character of the loads taken by the assembly depends on this. Rigid fastening to the test facility allows us to check the vibration strength of the assembly precisely under those conditions, which are set by the test program. With fastening which recreates the fastening of the valve on the engine the

assigned vibration conditions will exist only for a part of the fastening unit - the brackets installed on the test bench table - and the vibration conditions of the assembly itself will differ from the required conditions. Therefore, depending on the test purposes, this or another system for fastening the assemblies to the table is selected.

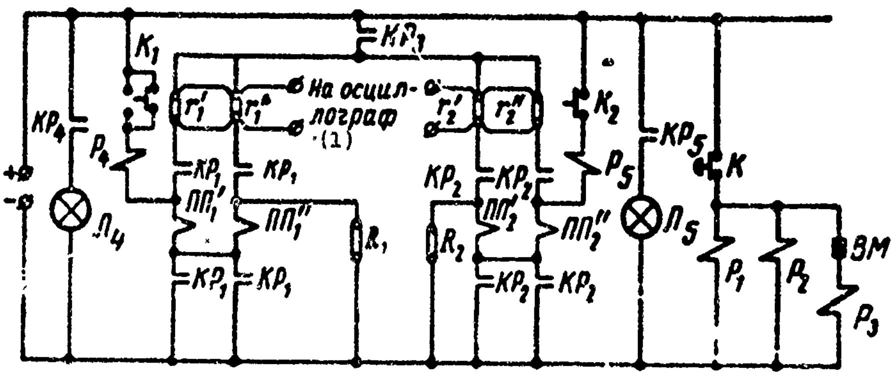

Fig. 7.6. Electrical circuit of the console: $\text{ПП}'_1$, $\text{ПП}''_1$ - circuits for heating the first explosive cartridge; $\text{ПП}'_2$, $\text{ПП}''_2$ - circuits for heating the second explosive cartridge; r'_1, r''_1; r'_2, r''_2 - resistances of the circuits of heating of the first and second explosive cartridges; P_1; P_2; P_3; P_4; P_5 - windings of the electromagnetic relays; KP_1; KP_2; KP_3; KP_4; KP_5 - normally-disconnected contacts of the appropriate electromagnetic relays; R_1; R_2 - resistances for checking the continuity of the circuits of the explosive cartridges with a safety current; K_1; K_2 - buttons for checking the continuity of the circuits of the explosive cartridges; K - start button; ВМ - timing mechanism, ensuring preliminary closing of the contacts of relays P_1 and P_2 relative to P_3; $Л_4$, $Л_5$ - control tubes. KEY: (1) to oscillograph.

It is essential that the construction of the table of the vibration stand be sufficiently rigid, since otherwise various points of the table will have different vibration overloads. This leads to errors in determining the actual value of the load acting on the assembly.

7.3. BASIC SAFETY MEASURES IN TESTING VALVES

The conducting of tests of valves for LPREs involves a number of dangers: the possibility of destruction of the assemblies, containers for tubes, which are under pressure of compressed gas, the danger of explosion of certain propellant components, the toxicity of some, the easy flammability of others, the high level of high-frequency noises and so forth.

Therefore, organizational and technical measures must be taken to exclude the possibility of explosions or failures, measures to ensure the safety of service personnel, and to protect workers from the effect of toxic substances and noise.

These measures must be directed to the mechanization of the processes of filling and emptying of components, the provision for fire-fighting preparedness, mechanization of assemblage of units on the test bench, the use of remote methods of controlling the test process and monitoring the hermeticity, eliminating the need to have people around the tested valves, which are under pressure, and so forth.

The safety technology service is occupied with the development of such measurements - both organizational and technical. The rules for safety technology when testing LPRE valves basically encompass:

a) handling with pneumatic test benches and pneumatic facilities;

b) operation of the hydraulic facilities for static and dynamic valve tests;

c) operation of the vibration test benches and vibration facilities;

d) handling aggressive toxic propellant components.

The safety technology goals for handling propellant components: provide for the sequence of conducting tests and rules for storing propellants, and also methods for handling them; establish the allowable concentration of propellant vapors in the atmosphere and in the workers' bays; point out the necessity for ventilation of the work site, as well as for neutralization of the drainage wastes and run-off residues of the propellant constituents; delineate the sequence for degreasing of reservoirs, conduits, and test bench assemblies; define methods of individual protection - type of overalls, brand of masks; and cite methods of first aid for accidents and so forth.

Similar conditions are also specified for working without a component - there is an established order for storage, assemblage and operation of the explosive cartridges, established methods for calculating strength and methods of checking the armored cabins, armored protective devices, individual and collective protection measures for high frequency noise, and there is specified the periodicity of protective and overall operations and so forth.

For ensuring work safety on the test benches the organizational side of the thing itself is of great, if not decisive, importance. For each test bench or group of test benches there must be designated a responsible person - most frequently, an experienced mechanic. On each stand there should be displayed the control system for it, where the number of valves, reducers, EPVs in the system must coincide with the marking of these assemblies on the stand itself. The levers of the most critical test-bench assemblies and of the emergency shut-off assemblies are marked with a special color. Each facility must have instructions for operation, which have been studied by all the workers on this facility.

A strict order must be established for the storage and consideration of the test bench assemblies, the dismantlable

tubing and equipment - the adapters, the plugs, branch tubes and so forth; in our view, this is one of the basic problems of ensuring fast and safe operation on the test benches. All the test bench assemblies, conduits and equipment must have technical documentation or the certification of a technical check. Routinely, at regular intervals, preventive maintenance of the test bench equipment, replacement of the instruments, and the inspection of the assemblies must be made.

The preparation for testing on the test bench must be made under the leadership of a single responsible person, with strict delineation of the problems and responsibilities of all the participants of the operation. Before the start of the tests the leader should have a clear idea of the purposes and of the program of operations. He should foresee the possible defects of the tested assembly, their consequences and measures to quickly eliminate these consequences.

Before the start of the critical tests a report is filled out for the preparedness of the facility for work, which is signed for the completion of readiness for the test by the persons responsible for this or another range of problems: responsible for the preparedness of the electrical system of the test bench, for the preparation of the measurement means, for the quality of the installation of the valve on the test bench, for the checking of the test-bench main lines, for the quantity of the supplied component, for its quality (the results of chemical analysis), for the preparedness of the fire-extinguishing system and so forth. Only after the filling out of such a report and its confirmation by the appropriate leader can one proceed to the tests themselves.

Such a procedure ensures not only working safety, but also a high degree of reliability of the obtained results.

Before the start of the tests, which involve the discharge into the atmosphere of a stream of air or water, audiovisual warning signals are given.

Any operations connected with the use of high pressure, toxic, explosion-dangerous or combustible components, explosive cartridges, must be carried out in the presence of no less than two workers - for the possibility of rendering help, if it is required.

These general statements are extended to all forms of tests of LPRE assemblies. Certain safety measures for specific forms of tests are presented below.

7.3.1. Working with Compressed Gas

All valves subjected to tests must be located behind a reliable enclosure, which protects the testers in case of failure of the tested valve. Such an enclosure may be a reinforced-concrete box, an armored cabin, closet or other armored-protective equipment - depending on the size of the valve and the pressure employed during the tests. Control of the tested device should be from outside the armored-protective device, excluding the necessity for a person to be there. When there is high pressure in the tested valve, the possibility of a person's being in the armored cabin must be excluded. This is achieved by the use of a pneumatic lock, which locks the door when pressure is supplied to the valve and prevents the supplying of the pressure when the door is open.

A maximum number of test bench assemblies must be located inside the cabin or in the special shielded panels. Assemblies which are located within the cabin must be protected by an armored sheet in case of failure of the tested valve. Test-bench assemblies located outside the cabin, where the servicing personnel are located, must possess especially high strength and reliability.

With the purpose of reducing noise during pneumatic tests the drainage tubes should lead out from the site to the street and mufflers of one sort or another should be installed there. The tubes themselves should be sound-insulated on the outside.

All the tanks and containers, which are not part of the tested device, if under pressure, must conform to the rules of Gosgortekhnadzor (State Committee of the Council of Ministers for Supervision of Industrial Safety and for Mining Inspection) [12], independent of whether these containers are located inside or outside the box; in case the product of the working pressure in the container p (in at) for the volume of the vessel V (in l) for air at a temperature $t \leq 200°C$ exceeds 5000 ($pV \geq 5000$), such a container must be officially registered with the inspection of Gosgortekhnadzor.

In making the armor-shielded devices special attention should be paid to the strength of the viewing windows, which are provided with bullet-proof glass; the use of a multi-layer glass laminate can be recommended (Fig. 7.7).

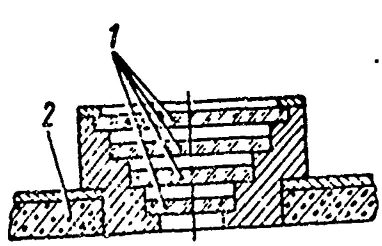

Fig. 7.7. Viewing window of the armored cabin: 1 - bullet proof glass; 2 - armored cabin wall.

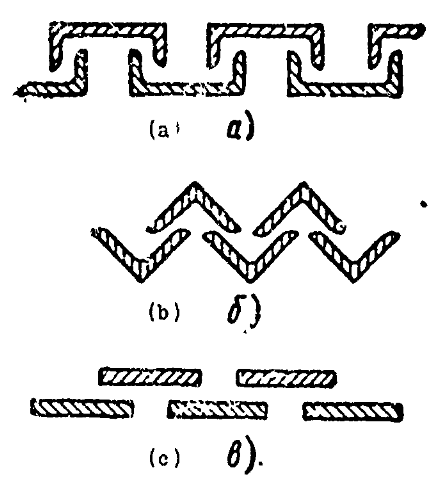

Fig. 7.8. Cross-sections of coverings of the armored cabins: a - made from channel iron; b - made from angle iron; c - made from strips.

Each armored cabin should have a covering, through which air could escape, but which would be impenetrable to fragments. Figure 7.8 shows cross sections of several of such coverings. The flow passage cross sectional area for air flow through the covering must be adequate, so as to avoid the possibility of increasing the pressure in the armored cabin during an explosion of the tested valve. The armored cabin must be reinforced by a foundation or, what is more advisable, it should be equipped with a floor made from a steel sheet, welded to the side walls.

Before the armor-shielded devices are put into use, they must be subjected to a special check for strength by the pneumatic destruction of the corresponding valve or an imitation of it. (With the manufacture of several samples of single-type devices one of these is subjected to testing). If different assemblies are to be tested in the armor-shielded device, then the one, for which the value of the product pV is maximum, is subjected to destruction. Destruction of such valves can be carried out with a test pressure $p_{исп}$ (p_{test}), which exceeds by 20% the working pressure p_p ($p_{исп} = 1.2\ p_p$), which creates a certain safety factor. Destruction of the assembly at a pressure exceeding the working pressure by a total of 20% is taken care of by a special weakening of the assembly. To make just such a weakening, which is necessary, is rather difficult, and a number of attempts must be made for this. Therefore, we can therefore recommend a design imitation of V. V. Sopolev, which guarantees the unjoining of the parts with the given pressure and permits us to make repeated tests without replacing the simulator.

The design of the simulator is shown in Fig. 7.9. Into the inner cavity of the simulator through connector 5 a certain pressure $p_{исп}$ is supplied. An auxiliary pressure (50-100 at) is supplied through EPV into the controlling cavity through connector 4; under the effect of this pressure ring 3 slips, balls 2 disintegrate,

housing 1 and cover 6 fly off[1] under the pressure $p_{исп}$. The advantage of such a construction consists also in the fact that the site of impact of housing 1 against the armor-shielded device can be preselected, i.e., the equipment construction can be tested at its weakest point (only the bulletproof glass is protected during the tests).

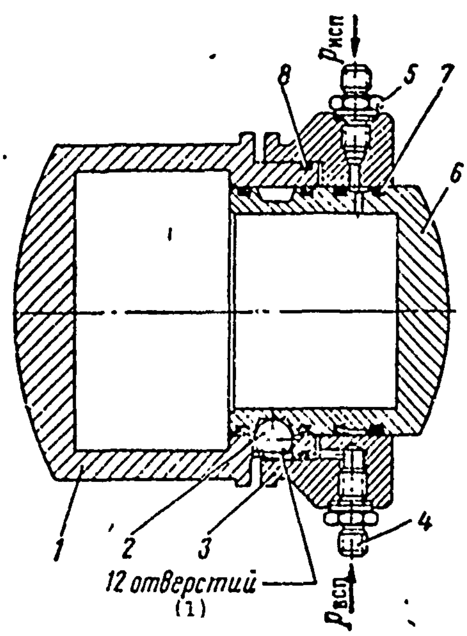

Fig. 7.9. Simulator of an assembly for checking the strength of an armor-clad unit: 1 - housing; 2 - ball; 3 - ring; 4, 5 - fittings; 6 - cover; 7, 8 - seal rings.

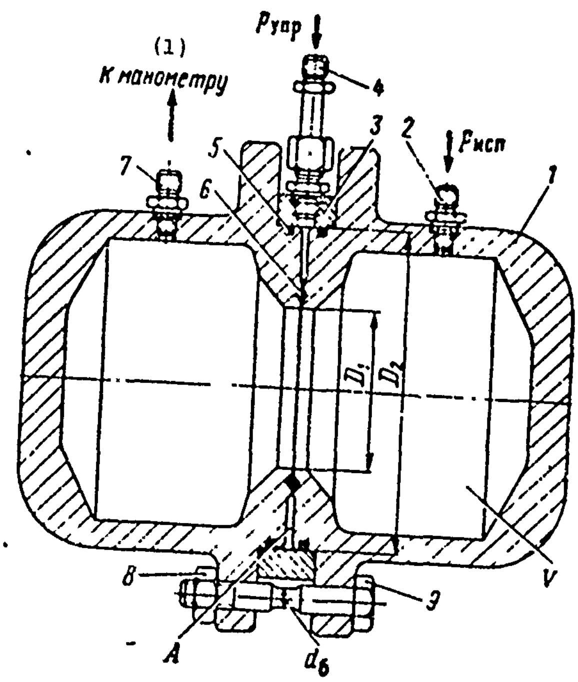

Fig. 7.10. Diagram of the valve simulator: 1 - half of the housing; 2, 4, 7 - connectors; 3 - ring; 5, 6 - rubber seal rings; 8 - nut; 9 - explosive bolt.
KEY: (1) To manometer

[1]The EPV is controlled remotely; during the tests of the armor-shielded device there must be no personnel in the test box.

Another original design of a valve simulator, suggested also by V. V. Sobolev (Fig. 7.10), is less universal, but on the other hand simpler to manufacture and more reliable. A diagram of the pressure feed to the simulator is demonstrated in Fig. 7.11).

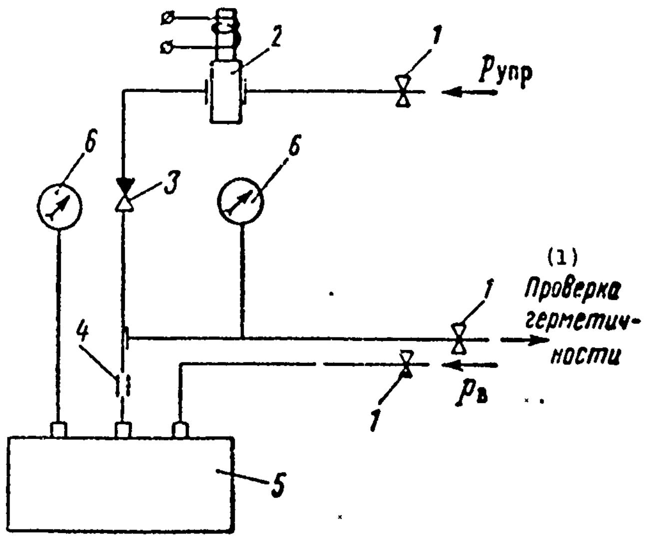

Fig. 7.11. Diagram of the pressure feed to the simulator: 1 - valves; 2 - electropneumatic valve; 3 - relief valve; 4 - discharge jet; 5 - simulator; 6 - manometers.
KEY: (1) Test for hermeticity.

To the inner cavity of the simulator through connector 2 (see Fig. 7.10) compressed air of a certain pressure $p_{исп}$ is fed. Bolts 9 are designed in such a way, that the test pressure $p_{исп}$, acting on the area of the circle with diameter D_1, produces neither the destruction, nor significant elongation of the bolts.

Then through connector 4 control pressure is supplied to the small cavity A. The force acting on the destruction of the bolts sharply increases as a result of the increase in the area, on which the pressure acts. Depending on the value of the controlling pressure, the bolts either immediately break, or (with low $p_{упр}$) ($p_{control}$) are stretched; with the stretching of the bolts the

hermeticity of rubber seal 6 is destroyed, and pressure $p_{исп}$ spreads into cavity A through discharge 4 (see Fig. 7.11), after which the bolts break, because pressure $p_{исп}$ will now act on the area of the circle with diameter D_2. Depending on the value of $p_{исп}$, the diameter of the neck of bolts $d_б$ is calculated (see Fig. 7.10).

Relief valve 3 (see Fig. 7.11) serves to prevent the spread of pressure $p_{исп}$ to EPV 2 in the case of sudden destruction of ring 6 (see Fig. 7.10) in the simulator.

To reduce the volume V, the cavity of the simulator can be filled with metal balls, water, etc.

This is a repeat-action simulator, since after the replacement of the bolts it is suitable for repeated use.

As a result of the tests the armor-shielded device should not be subjected to failures or should not have such external damages, which could be dangerous for service personnel (the tearing out of the external bolts, nuts, connectors, and so forth).

Pneumatic assemblies for testing assemblies which have a value $pV \leq 3000$ at·l are allowed to be placed inside industrial buildings; with values of $pV > 3000$, but less than ≤ 25000 insulated rooms are necessary (if $p > 6$ at). Devices which operate at values of $pV > 25000$ at·l can be situated only at the test stations, removed from other working sites.

To reduce the danger of the hydraulic tests attempts should be made to reduce the volume, which is filled with compressed air. Thus, in checking the hermeticity of welded seams of a housing of large volume, the housing can be filled with a special filler made from metal or plastic, occuping a large part of

the inner volume of the housing. Such a method permits pneumatic tests of parts of large assemblies to be conducted in comparatively weakly reinforced cabins.

To create a liquid cooling medium during valve tests at low temperature freon is frequently used; dry ice (solid carbon dioxide) is added to it. In the process of dissolving the carbon dioxide in the freon intensive gas liberation takes place; with a freon concentration in air equal to 4 g/l a strong, pungent odor can be perceived. Therefore, the freon baths must be covered and have collectors from the exhaust fan system on the side. After reaching a certain temperature, when the dry ice is dissolved, the smell of the freon is almost imperceptible.

In the presence of an open fire freon decomposes with the formation of highly toxic substances.

7.3.2. The Installation and Operation of Conduits

In the creation and operation of test-bench devices, consoles, boxes for pneumatic tests and tests with propellant, great attention should be paid to the installation of the conduits which operate under high pressure (especially conduits with an internal diameter of 10 mm and higher), and to handling them. Conduits should be manufactured, installed and operated in conformance with the effective rules (see, for example, "Rules for Installation and Safe Operation of Air Compressors and Air Conduits," Handbook on Safety and Industrial Sanitation Procedures, Publishing House "Sudostroenie," Leningrad, 1965).

With an internal diameter of the conduit exceeding 15 mm, in order to check the quality of the welded seam, monitoring by x-rays or gamma-rays is used (according to GOST 7512-55). Sometimes nonferrous defectoscopy ultrasonic monitoring or

magnafluxing is used. Not less than 5% of the turning joints (i.e., those joints, during the welding of which the welder was able to turn the tubes) and all 100% of the stationary joints (where the tubes were not turned during welding, and the welder who was sometimes in a position uncomfortable for working, penetrated the entire perimeter of the joint).

Especially careful control should be exercised on the state and operation of flexible rubber hoses. They must be manufactured according to the appropriate approved plans. With the reliable sealing off of the ends the hoses, cited in GOST 6286-60, can be employed at pressures of 90-300 at (depending on the internal diameter).

The indicated rubber hoses are designed for operation using compressed air or other nonaggressive gas. Such hoses undergo visual inspection every two months; they undergo hydraulic pressing at 150% working pressure every six months.

7.3.3. Conducting Static and Dynamic Hydraulic Tests

In comparison with pneumatic tests, hydraulic tests present less danger. However, the basic rules for conducting pneumatic tests (and the rules for monitoring the conduits) are also extended to hydraulic tests, especially at high pressures.

When conducting distructive tests one should see to it that the liquid completely fills the entire inner volume of the valve, without permitting the formation of air pockets. Therefore, with a complex configuration of the internal cavities of the assembly it is best to pour water under a vacuum, i.e., to first evacuate the inner cavities of the valve.

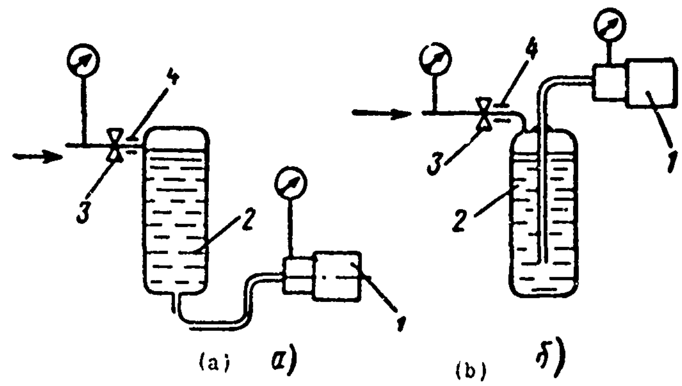

Fig. 7.12. Layouts for testing an assembly for strength (liquid pressure is created by compressed gas): 1 - tested assembly; 2 - cylinder; 3 - valve; 4 - discharged jet.

To create liquid pressure, one should use a hydraulic press, without permitting charging by compressed air for this purpose: during deformations of the housing of the valve gas will enter inside and in the case of destruction of the housing can lead to serious consequences.

However, sometimes circumstances force us to use just such pressure charging when testing by hydraulic destruction. In this case, too, it is essential to ensure the impossibility of entry of the gas into the tested object by employing a buffer cylinder of known strength, connected to a discharge tube and a conduit with the tested object (Fig. 7.12); a layout according to the type shown in Fig. 7.12b is preferable.

7.3.4. Tests for Vibration Stability and Strength

Vibration tests with gas pressure in the assembly can be conducted only in an armored cabin, corresponding to the above cited condition; the equipment should also have noise protection. With this aim in mind, sound insulation of the armored cabin with plates made from mineral betting and sheets of polyurethane

(but this is dangerous with respect to fires) is used. According to literature data, sound absorption is also achieved by putting on the walls of the armored cabin special varnishes with a small layer thickness.

To reduce noise acoustic explosions should be made on the foundations of the vibration stands and buffeting stands. Moreover, service personnel should use individual protective measures - special ear protectors, ultrathin cotton or noise-absorbing inserts for the ears.

Simulators can be installed in place of the working explosive cartridges during vibration tests of the explosive automatic systems. Here the hermeticity of the installation of explosive cartridges and the rigidity of the fastening site will be checked. However, in order for a simulator to be used, one must have confidence in the vibration stability of the explosive cartridge itself.

Tests of explosive cartridges can be made by placing the explosive cartridge into a special device, designed and made for detonating explosive cartridges of significantly greater power.

7.3.5. Working with Propellant Components

In creating test benches for testing valves with real propellants the entire operating test bench system must be carefully thought out, beginning with the supplying of propellant to the stand, storing it and finishing with the removal of substandard products. Any unfinished operation can result in a serious accident.

Roads for transporting the propellant must be smooth, without pits or bumps or depressions; the transportation of propellants is permitted only in vehicles specially designed for this purpose.

The storage for various types of fuel should be organized in various places. It is not allowable to store fuel and oxidizer together. Such an environment inevitably leads to fire or explosion. The temperature in the storehouse should not exceed what is permitted for the given product. Therefore, the possibility of air conditioning should be provided for.

It is not permitted to conduct tests one after the other, first with an oxidizer, and then with fuel, in one and the same test box, and with the same equipment. The box must be adapted for the tests of the designated product, it must have the proper ventilating equipment and collectors, placed either above, or below - depending on the density of the propellant vapors. An independent drainage channelization system should be provided for, separated from the common sewage system; the unification of the channelization systems with the common sewage system, even if it does not lead to an accident (although this is entirely possible), will, in any case, cause inadmissible contamination of the waste waters.

It is forbidden to conduct welding operations at the work sites, where propellant is found. The walls of the boxes and the equipment should be covered by a special varnish, which prevent the introduction of propellant vapors into the pores of the walls. This greatly improves the working conditions, and allows us to easily clean the walls and equipment, and also reduces the concentration of propellant vapors at the site of the box.

Conducting tests with a heated propellant with a low boiling temperature under pressure, it is necessary to consider that the pressure differential will lead to gasification. Therefore, it is not permitted to install a discharge jet, washer or other hydraulic resistance on the mainline of the fuel product in front of the drainage tank. This will lead to gasification, discharge

of vapors through the drainage outlets to the atmosphere, i.e., to the gas contamination of the environment. It is therefore necessary to install a cooler in front of the discharge jet, so that the pressure drop occurs on the cold product. Nonhermeticity of tubing with hot propellant is very much to be feared, because this leads to intensive gasification in the box.

When working with such an oxidizer as liquid oxygen, complete purity is the most important requirement for safety technology. The air which goes for pressure charging of the liquid oxygen containers must not contain traces of oil. The tool used on the work bench must be copper coated to avoid the formation of sparks, which can lead to an explosion. From considerations of explosion safety it is not permissible to cover the floor of the box with asphalt - it should be cemented.

While working at the test bench, it is forbidden to wear silk, synthetic or woolen underclothing.

Working with hydrogen peroxide - a very explosive-hazardous propellant - requires a high degree of purity. Catalysts for it are (among other products) ferric oxide and sand. All the internal surfaces of the containers, tubes, test-bench assemblies require careful cleaning and passivation. The danger of explosion increases especially with the increase in the peroxide concentration. This propellant decays violently with an increase in temperature. Strict observation of the temperature of the product, and the taking of measures to prevent a temperature rise exceeding 25°C is one of the first cares of the testers.

CHAPTER 8

ENSURING OPERATING STABILITY OF PROPELLANT VALVES

In examining the autonomous requirements imposed on propellant valves, it was cited that one of the most important requirements is the assurance of operating stability of the assembly.

The degree of stability is characterized by:

a) the value of non-coincidence of the opening times (closing times) of various samples of a valve of one and the same type under the same conditions, which is the result of individual peculiarities of manufacture of explosive cartridges, springs, seal rings, clearance value, quantity of lubrication and so forth;

b) value of non-coincidence of the opening times (closing times) of one and the same sample of a valve under different conditions possible during operation, i.e., at different temperature conditions, different inlet and controlling pressures;

c) the value of non-coincidence of the opening times (closing times) of one and the same sample of a valve under the same conditions, which is the result of frictional instability.

General instability of valve operating (triggering) times of one and the same type valve is caused by the combination of all the above mentioned time variances.

Although the value of instability of operation of explosive valves according to paragraphs b and c cannot be checked directly, it is known that the highest stability is possessed by direct-action explosive valves, whose triggering time variance does not exceed 0.003 sec. The operating stability of check-type explosive valves is determined by the operation of the hydraulic-brake device. With a perfected hydraulic-brake design the instability of this type of assembly does not exceed 0.01-0.03 sec.

The greatest difficulties ensuring operating stability are encountered in pneumatically controlled repeat-action valves. Their triggering time is one order higher than the triggering time of explosive valves. It is natural to expect that the greater the absolute triggering time, the greater will also be the absolute variance of these times, i.e., the greater the instability.

Examined below are problems connected with providing stability for pneumatically controlled valves.

8.1. STABILITY OF PNEUMATIC VALVES FOR HIGH-BOILING LIQUIDS

Let us examine the process of opening at normal temperature for a test-bench valve, shown in Fig. 8.1. A study of the valve's opening times was made on a pneumatic device, the diagram of which is shown in Fig. 8.2. Compressed air was supplied to the inlet to the liquid cavity of the valve instead of propellant; the pressure at the inlet was maintained constant.

In the initial position the valve is closed by the controlling pressure in cavity A (see Fig. 8.1). To open the valve an electrical signal is supplied to the switch of normally open EPV 2 (see Fig. 8.2) with drainage, as a result of which air is leaked from the controlling cavity and under the effect of the inlet pressure and the spring the valve opens. The process of opening

is examined, by taking the giving of the electrical signal[1] to the EPV as the starting moment. At the outlet of the pneumatic valve washer 3 is installed, with the aid of which the air flow rate is determined.

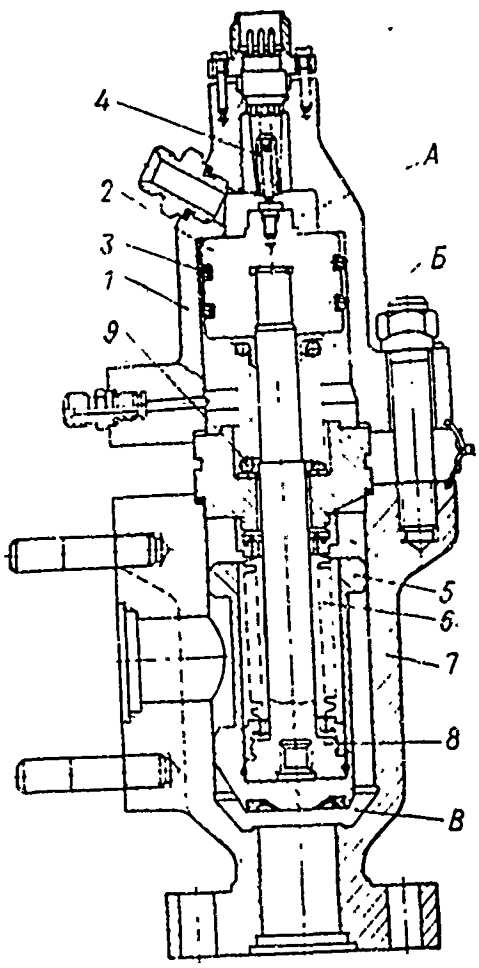

Fig. 8.1. A normally open test bench pneumatic valve: 1 - cover; 2 - piston; 3 - seal ring; 4 - valve travel signal indicator; 5 - valve; 6 - bellows; 7 - housing; 8 - rod; 9 - spring; A - control cavity; Б - drainage cavity; B - propellant cavity.

[1]The command to the switch of the EPV is transmitted through an intermediate relay. For simplicity they are not shown on the diagram; we also disregard their triggering time.

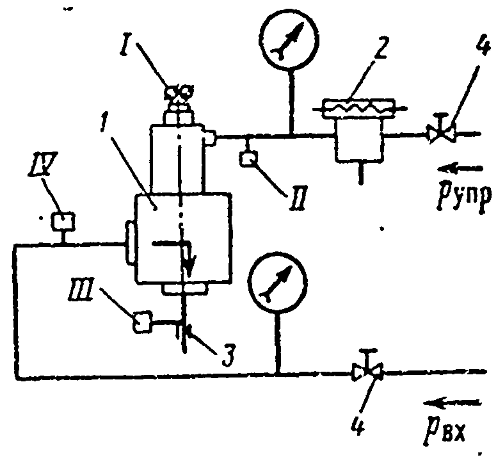

Fig. 8.2. Diagram of a s t up for testing the pneumatic valve: 1 - tested pneumatic valve; 2 - normally open EPV with drainage; 3 - flow-rate washer; 4 - shut off valve; I - moving system travel sensor; II - controlling pressure sensor; III - valve outlet pressure sensor; IV - valve inlet pressure sensor.

The recordings of four sensors are shown on the schematic cyclogram of the triggerings of such a pneumatic valve (Fig. 8.3):

- sensor I, fixing the pressure (travel h) of the moving system of the valve;
- sensor II, fixing the controlling pressure $p_{упр}$;
- sensor III, fixing the pressure at the outlet from the pneumatic valve $p_{вых}$;
- sensor IV, fixing the pressure at the inlet $p_{вх}$.

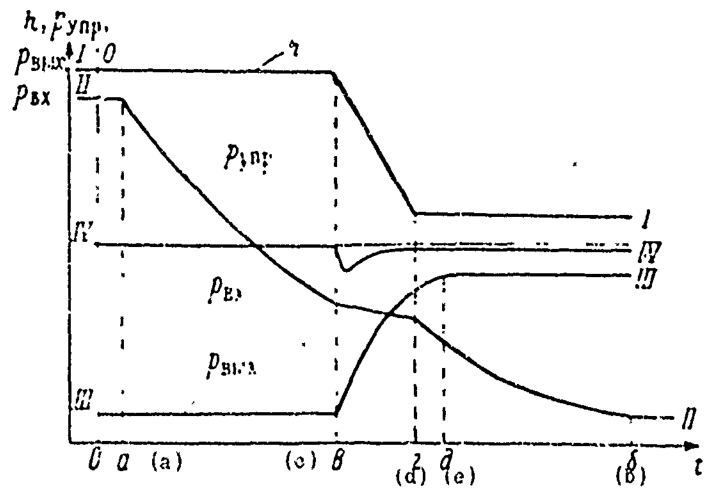

Fig. 8.3. Diagram of the change in parameters during the opening of the pneumatic valve: I - travel sensor indicators; II - change in controlling pressure; III - change in outlet pressure; IV - change in inlet pressure.

The time computing origin - the moment of the command to the EPV - is the vertical straight line 0 - 0. The segment 0 - c is the time of delay for the start of movement of the pneumatic valve's moving system. The segment c - d - is the time of movement of the moving system (on the cyclogram travel h is depicted by a straight line). The segment 0 - d is the time of complete opening of the valve.

During time 0 - a the displacement of the moving system of the EPV and the rarefaction wave propagation takes place along the tubing to the controlling cavity of the pneumatic valve. The triggering time of the relief-type EPV (segment 0 - a), depending on the construction of the EPV, supply voltage, temperature and so forth varies within the limits of 0.02-0.06 sec.

At moment of time a the discharge of controlling pressure p_{ynp} begins and at moment e it falls to 0 at moment c, when the pressure p_{ynp} was reduced (it has not yet fallen to zero), the shifting of the moving system begins under the action of pressure p_{BX} and the force of spring 9 (see Fig. 8.1). Pressure p_{BbIX} begins to increase practically simultaneously with the start of movement of the valve's moving system. The achievement of steady pressure at the outlet (point e) can take place both after and before the finishing of movement of the valve - this depends on the flow rate magnitude through the system (on the diameter of flow-rate washer 3, see Fig. 8.2) and on the value of the volume between the valve seat and the flow rate washer. With the start of movement of the moving system, as a result of the decrease in volume of the controlling cavity, the drop in the controlling pressure ceases either almost completely, or is abruptly retarded (segment c - d in Fig. 8.3). The discharge rate of the controlling air even influences the time of shifting of the valve, but insignificantly.

The moment of the start of movement is unstable, i.e., the value of the delay of the start of movement - segment 0 - c. The value of the delay is basically determined by the rate of leakage of controlling air pressure.

The time of the drop in the controlling pressure is determined by many factors:

- the volume of the controlling cavity the smaller this volume is, the more quickly the pressure falls;
- the flow passage cross sectional areas of the EPV; the greater they are, the less the leakage is;
- the length and diameter of the tubing between the EPV and the pneumatic valve; the less its length is and the greater its diameter, the more quickly the pressure falls (true, an increase in diameter is equivalent to an increase in volume of the controlling cavity);
- the value of the controlling pressure; with a decrease in the pressure the time of discharge is reduced (however this influence within the limits of 5-10 at is not very significant);
- the temperature of the controlling gas; the lower the temperature, the higher the gas density, and the time of discharge increases; this phenomenon is most clearly noticeable when handling low-boiling propellants (see below);
- the kind of controlling gas; thus, the time of discharge of compressed air is several (1.3-2.5) times greater, than the time of discharge of helium at the same pressure;
- the pressure of the propellant; its influence is determined by the valve design.

The moment of the start of movement is based not only on the rate of air discharge[1], but also on the value of the propellant

[1]The time of opening of a normally closed valve is influenced by the rate of entrance into the controlling cavity; the discharge of the gas determines the time of valve closing.

pressure, the pressure of the springs and, what is very important, the shear forces of the rubber rings (or seal rings). Depending on the value of the friction in the seal, with all other conditions being equal, movement may begin earlier or later, i.e., corresponding to a larger or smaller controlling pressure.

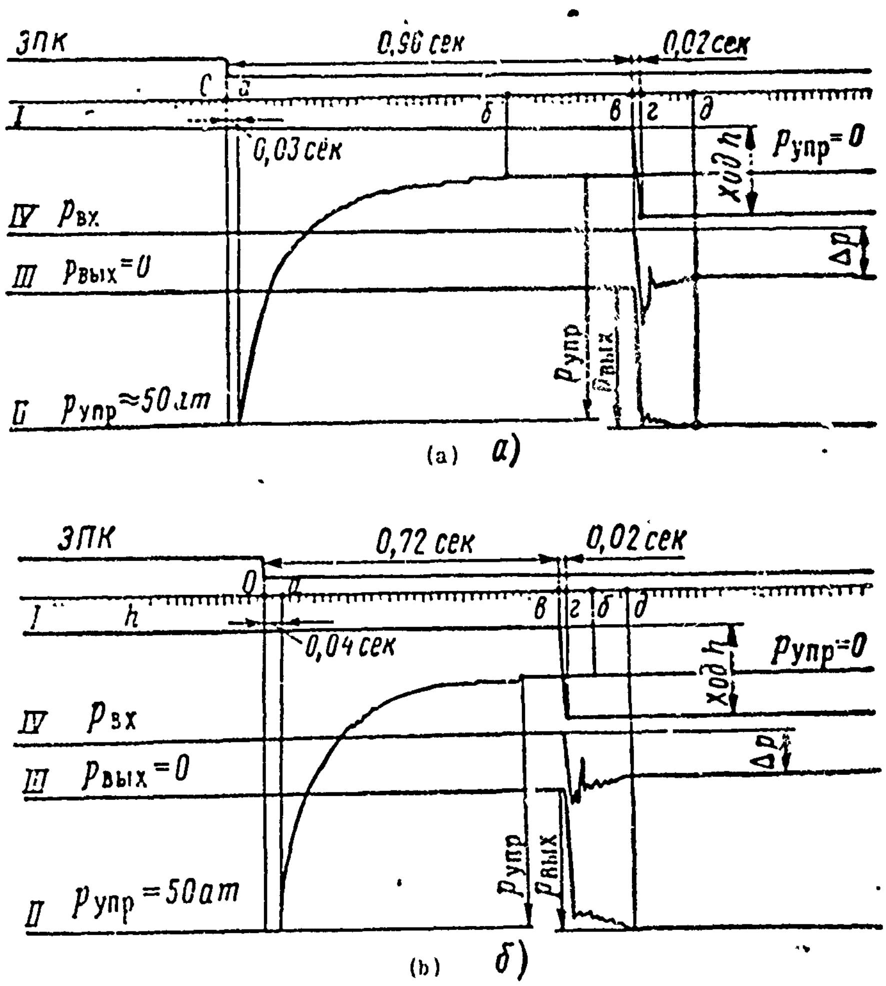

Fig. 8.4. Oscillograms of valve opening: a - first triggering after holdi g under controlling pressure for 2 hours; b - repeat d triggering after 30-40 sec after the first one (Designations are the same as in Fig. 8.3).

With the first triggering (after holding under the controlling pressure for two hours) movement began, after the controlling pressure was reduced to zero (see Fig. 8.4a). The beginning of movement was determined not by forces, subject to calculation, but by the friction of the rings. The second triggering (see Fig. 8.4b) occurred 30-40 seconds after the first. The delay of movement was reduced by 0.24 sec. Instability is clearly present here.

To reduce the triggering instability the frictional force should be reduced. Thus, the installation of rings with less compression reduces the force of elasticity of the rubber, and at low pressures reduces also the very value of the shear forces, and its variance. In all these cases the resulting force acting on the valve's opening is increased. Especially noticeable is the influence of compression at a temperature of -40°C, when the hardness of the rubber is increased.

The effect of the degree of normalization of the rubber affects the value of friction of rubber seals, especially at low temperatures. Certain brands of rubber possess the ability to absorb the silicone liquid, which is a constituent part of the lubricant greases. If a newly manufactured part made from such rubber is coated with lubricant containing silicone liquid, after a certain time the surface of the part becomes dry - the rubber absorbs the lubricant; the force of friction of such a part (seal ring or regular ring), in comparison with a lubricated part, abruptly increases. In order to eliminate this influence, after manufacture normalization of the parts is carried out, by immersing them for several days into silicone liquid (until the weight of the parts ceases to change due to the absorption of the lubricant).

Table 8.1 presents data which characterize the influence of temperature, percentage of compression, normalization, type of

rubber and time at rest on the shear forces of rubber rings with an external diameter of 30 mm (diameter of the cross section is 3 mm). The tests were conducted on five valve simulators, shown in Fig. 8.5.

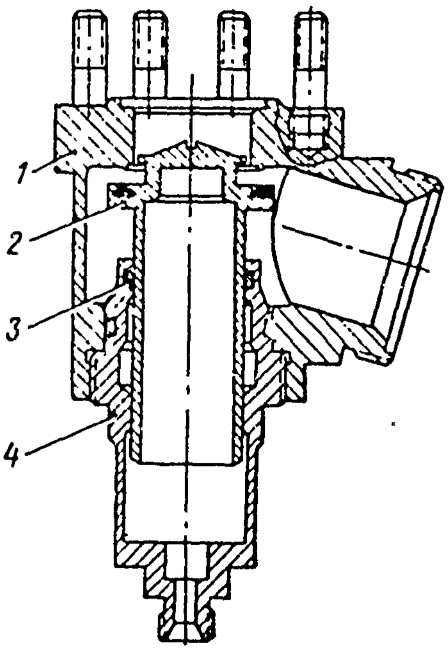

Fig. 8.5. Valve simulator, designed for testing friction of seal rings: 1 - housing; 2 - valve; 3 - tested rubber ring; 4 - cover.

The obtained results testified to the increase in friction forces with a reduction in temperature from +20°C to -40°C, which corresponds with the conclusions made on the basis of numerous tests of various pneumatic valves.

The test results showed that when evaluating the working efficiency of a valve one should take into consideration not the average data from several triggerings in succession, but the results of only the first triggerings.

However, apart from the influence of relatively short-term holding in the stationary state, connected with the extrusion of the lubricant, there is yet another phenomenon exhibited: "adhesion" of the rubber, caused by long-term holding in the stationary state.

Table 8.1.

Type of rubber	% of compression of the rings	Degree of normalization	Test temperature	Forces in at	
				first movement after a 24-hour interruption	second movement 30-50 sec after the first
Based on natural rubber SKS-30	19-20	Series rings	15-25°C	0.4-0.54	0.27
Based on natural rubber SKB	17-22 17-22	Series rings	15-25°C -40 to -45°C	0.62-0.7 0.42-3.7	0.3-0.4 0.3-0.6
Based on natural rubber SKS-30	9 17-22	Normalized	-40 to -45°C	0.3-0.4 1.2-2.0	0.2 0.8-1.2
	17-22 9	Normalized under special observation		0.7 0.3	0.25 0.2

Experience has shown that after rubber has remained in the clamped state for a six-month period (clamped to steel or anodized aluminum alloys) adhesion of the rubber to the metal takes place. This phenomenon is especially noticeable when checking at low temperature, after prolonged holding at elevated temperature.

Figure 8.6 shows the change in the coefficients of friction of smooth and rough rubber on steel depending on the time of the interruption in movement, obtained by the American D. F. Denni.

Everything said above also refers to the operation of rubber seal rings. The shear forces in seal rings are even greater than in regular rings.

The assembly depicted in Fig. 8.7 was tested at temperatures of +50°C to -40°C with various seal rings. The pressure required for the beginning of movement of rod 2 with air pressure supplied to the connector of bushing 8 was determined. Tests were conducted on five samples of valves, each time waiting three days after assemblage. The test results are shown in Table 8.2 and in Fig. 8.8.

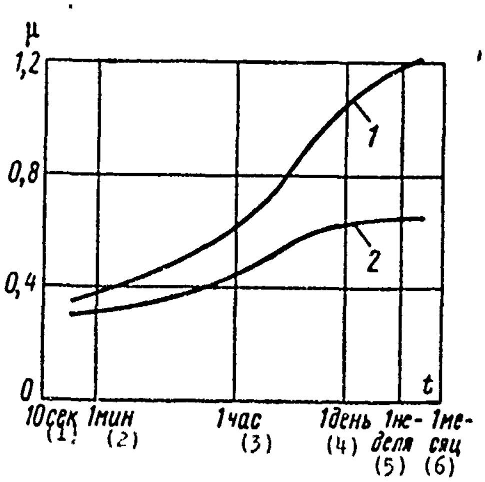

Fig. 8.6. The change in the static coefficient of friction of rubber on steel depending on the duration of the friction pair in the stationary state: 1 - smooth rubber; 2 - rough rubber.
KEY: (1) seconds; (2) minute; (3) hour; (4) day; (5) week; (6) month.

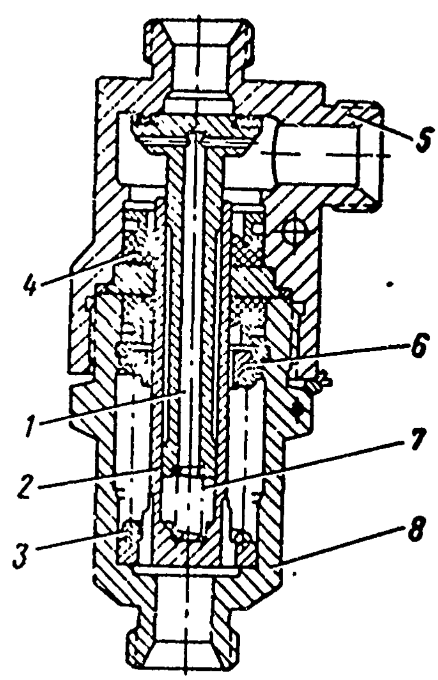

Fig. 8.7. Propellant valve: 1 - valve; 2 - rod; 3 - spring; 4 - seal ring; 5 - housing; 6 - spring detent; 7 - small spring; 8 - bushing.

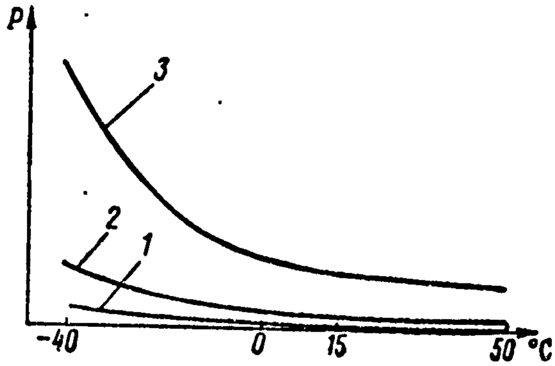

Fig. 8.8. Dependence of the shear forces of the seal rings on the temperature: 1 - seal rings made from rubber based on natural rubber SKB (specially normalized); 2 - series seal rings made from the same rubber; 3 - seal rings made from rubber based on natural rubber SKMS-10.

Table 8.2.

Test temperature	Contact pressure of the rod in at		
	series seal rings made from rubber based on natural rubber SKB	normalized[1] seal rings made from rubber based on natural SKB	non-normalized seal rings made from rubber based on natural rubber SKMS-10
15-20°C	17.5-20	14-18.5	30-41.5
+50-55°C	16-20	14-15.5	29-32
-40 to -45°C	30-85	18-22	100-120

[1]The seal rings were normalized under special observation.

From Table 8. the conclusion can be drawn that with a decrease in the temperature (with all other conditions being equal) there is an increase in the time interval from the moment of the electrical signal to the EPV until the start of movement of the moving system of the valve - as a result of the change in the controlling pressure, at which movement begins.

The stability of operation of the assembly will be adversely influenced by too much lubrication of the parts, when the freezing temperature of the lubricant is close to the lower limit of the operating temperature (furthermore, in this case the lubricant promotes the emergence of a leak in the seals).

The presence of jerks, jamming, delays in the shifting of the parts can have a very negative effect on the operating stability; therefore, in designs of these assemblies antifrictional electroplating of the metal parts is often employed.

Factors which determine the instability of frictional forces were examined above. However, there are limitations in the possibility of reducing the inconstancy of the frictional forces, then a more obvious means (but not always more acceptable) for increasing their stability is to reduce the time of controlling pressure discharge. Then, even if the value of the opening time variance was not changed in a percentage ratio, it will be unconditionally reduced in absolute calculation (in seconds).

The operating stability of assemblies, in which bellows are used, is significantly higher than in assemblies with seal rings or rings.

The operating stability of a valve depends in large degree on the rate of change in the propellant pressure at the valve inlet; this is determined by the engine layout and by the moment of the beginning of operation of the turbopump assembly (TPA) - up to or after the opening of the propellant valve.

If the TPA begins to operate earlier than the pneumatic opens, then the valve opening instability will also be determined by the character of the propellant pressure rise at the valve inlet, and by the very moment of the beginning of operation

of the turbine. In this case valve operating stability will be worse, i.e., the variance of triggering times may be greater[1].

The time of valve closing will always depend not only on the value of the air pressure in the controlling cavity and on the value of the propellant pressure, but also on the value of the pressure differential on the valve's locking mechanism.

To increase operating stability of pneumatic valves which handle high-boiling propellants we can recommend the following measures:

a) reduce the friction in the moving system - use bellows or (what is less effective) round rings with stop washers instead of seal rings; use rubber of the appropriate brands and high-quality normalization of it; provide adequate centering of the moving system;

b) increase the rate of discharge and feeding of the controlling gas by means of increasing the flow passage cross sectional areas and by reducing the length of the communication tubes for the compressed gas; reduce the volume of the controlling cavity; stabilize the controlling pressure; make a rational selection of the controlling gas itself (air, nitrogen, helium and so forth).

8.2. STABILITY OF PNEUMATIC VALVES FOR LOW-BOILING LIQUIDS

The operation of propellant valves with low-boiling liquids entails significant peculiarities in the conditions of ensuring triggering stability. Let us examine the operation of a valve handling liquid oxygen (see Fig. 2.9), even though everything

[1]For low-boiling propellants the situation is somewhat different, as will be shown below.

said below will also refer to operation with liquid nitrogen or other low-boiling liquids.

All the above expressed general considerations on the operating stability of valves also hold completely true for this valve, with the exception that instead of the friction forces in this case the rigidity of the bellows has an effect on the operating stability. However, in comparison with the variance of the shear forces of the seal rings or rings, the change in rigidity of the bellows with a change in external conditions is disregardably small.

The character of the gas discharge from the controlling cavity, which depends basically on the properties and on the state of the gas, has the greatest effect on the time of delay of the start of the valve's opening.

Research was conducted on equipment, the principal layout of which is similar to that shown in Fig. 8.2, except that in place of air at the valve inlet - into the propellant cavity - liquid oxygen was supplied in this case. After filling the liquid cavity with liquid oxygen the entire valve and, consequently the air in the controlling cavity, begins to be cooled.

Measures are taken so that the oxygen in the propellant valve will always be in the liquid, rather than in the gaseous state. Since the air in the controlling cavity is under a pressure of 40-50 at, the condensation temperature of this air will be higher than the temperature of liquid oxygen, having a pressure close to atmospheric.

The dependence of the compressed air condensation temperature on the pressure is shown in Fig. 8.9.

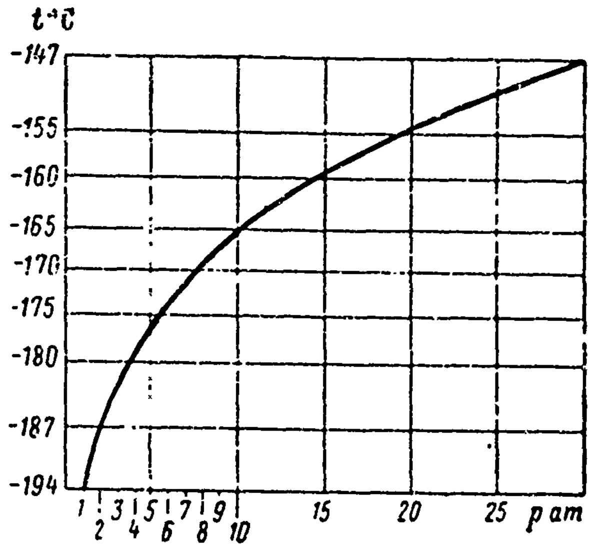

Fig. 8.9. Dependence of the compressed condensation temperature on the pressure.

As the valve housing cools, the compressed air will first change its density, and then begin to condense. We will note that initially in the construction of the valve shown in Fig. 2.9 the installation of a discharge jet in plug 8 was not provided for, and so the flow of air through the controlling cavity could not take place. The lower the oxygen temperature and, chiefly, the greater the time the oxygen valve exists in the medium of liquid oxygen (at a steady temperature), the greater the quantity of liquid air will accumulate in the controlling cavity, which is connected by tubing with the source of the compressed air supply. On command to open the air from the controlling cavity begins to discharge through EPV and the pressure of the controlling air falls; at some pressure, depending on the temperature, violent boiling (evaporation) of the liquefied air begins. For example, at a temperature of -169°C in the controlling cavity boiling begins at a pressure of 8 at, while at t = -154°C - at a pressure of 20 at (see Fig. 8.9). As a result of the boiling of air the pressure drop in the controlling cavity is retarded, and in certain instances for several hundredths of a second it even completely ceases. The process of air discharge is prolonged.

Thus, liquefaction of a gas in the controlling cavity during the holding of the valve under a component and the boiling of the gas during pressure discharge are characteristic peculiarities of an assembly handling low-boiling liquids.

During the discharge of air from the controlling cavity drops of liquid air - in one case more, in one case less - will also be drawn off by the flow. Consequently, even under identical conditions the process of opening will not be stable. The greater the escape of air in the liquid phase, the shorter will be the time of the pressure drop in the controlling cavity. The opening time depends greatly on the temperature in the controlling cavity - the lower the temperature, the greater the opening time. With identical temperatures the opening time will depend on the cooling time of the valve: the greater the cooling time, the greater the quantity of air manages to be condensed and the greater will be the time of the pressure release.

Figure 8.10 shows oscillograms of the actual process of opening of an oxygen valve: in Fig. 8.10a - during operation with water instead of oxygen, and in Fig. 8.10b and c - with the presence of oxygen in the liquid cavity, where in Fig. 8.10b valve opening occurred 15 minutes after filling with oxygen, and in Fig. 8.10c - after a two-hour holding of the valve under oxygen. Test results are shown further in Table 8.4 and in Fig. 8.11 - curve 2.

From Fig. 8.11 and Table 8.4 it follows that an increase in the valve's holding time under oxygen resulted in a pronounced increase in its opening delay. With a further increase in the holding time (more than two hours) the time of triggering of the valve was increased even more.

To reduce the instability, an uninterrupted flow of gas was introduced through the controlling cavity of the propellant valve and through the discharge jet in plug 8 (see Fig. 2.9).

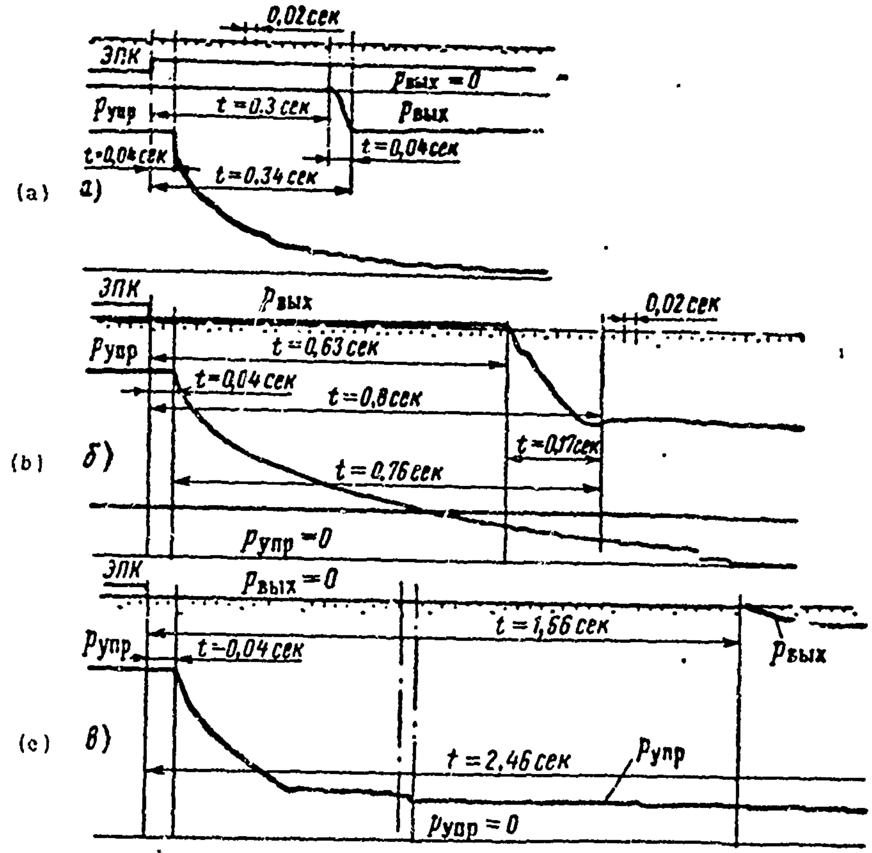

Fig. 8.10. Oscillograms of the process of opening of an oxygen valve: a - operating with water; b - after holding under liquid oxygen for 15 minutes; c - after holding under liquid oxygen for two hours.

Because of the exchange of air in the valve the temperature in the controlling cavity was increased and the air began to liquefy to a lesser degree, which resulted in a decrease in both the triggering time, and in the value of the variance.

This construction requires additional air output; the output value determines the efficiency of the method.

The dependence of the delay time for the start of opening of such a valve on the time of its holding under liquid oxygen can be seen in Fig. 8.11, curve 1. As is evident from Fig. 8.11,

when holding the assembly under oxygen for more than two hours the time of the valve's opening is practically unchanged. For the sake of comparison the test results of a valve without an airflow through the controlling cavity (curve 2) are plotted by a broken line.

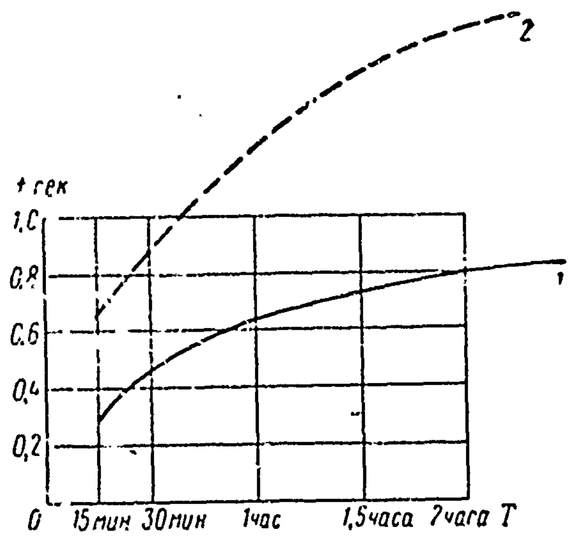

Fig. 8.11. The change in the delay of the start of valve opening (from the moment of the command to the EPV) as a function of time T of holding the valve under liquid oxygen: 1 - with the flow of air through the controlling cavity; 2 - without air flow through the controlling cavity.

Figure 8.12 shows the experimental curves for the change in the controlling pressure under various conditions and the moment of start of valve opening. The pressure in the controlling cavity at the moment of the start of valve opening when the valve is held under liquid oxygen (points A_{II} and A_{III}) is somewhat higher than when the valve is handling water, in connection with the fact that the spring force (acting on the opening side) increases with a reduction in temperature.

While the valve is held under oxygen the controlling pressure value itself exerts a significant influence on the time of the air release; the higher the pressure, the more liquid air is condensed at this temperature (because the condensation temperature is increased), and, consequently, the more the opening time is increased. Therefore, at a lower controlling pressure the process of opening will go faster and, consequently, more stably.

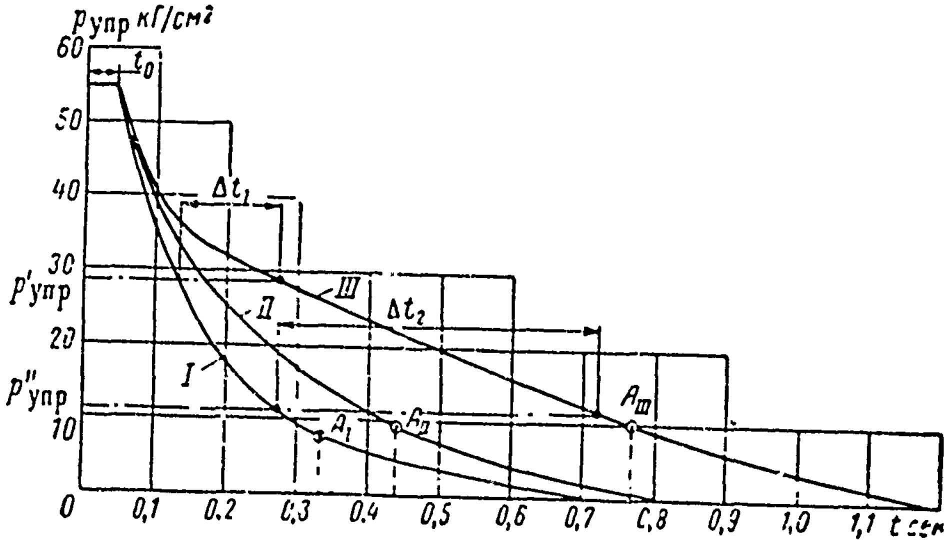

Fig. 8.12. Experimental curves of the change in pressure of the controlling air for an oxygen valve: I - operating with water; II - with a 30-minute holding of the valve under oxygen; III - with a 24-hour holding of the valve under oxygen; 0 - the moment of giving the command to the EPV; A_I, A_{II}, A_{III} - moments of the beginning of valve opening.

The experimental data indicate the effect of the value of the controlling pressure on the valve's opening time (see Table 8.3).

The valve's opening stability is basically influenced by the value of the pressure in the controlling cavity $p_{упр}$ at which it opens. The value $p_{упр}$ depends on the pressure of the oxygen in the liquid cavity, the spring force, the area of action of the pressure and so forth.

Table 8.3.

Controlling pressure kgf/cm^2	Time of opening of the valve, sec
55	0.40
35	0.31
25	0.22

The greater the pressure of the controlling cavity at the moment of opening, the higher the stability, since the process of evaporation of the liquefied air will occur after the valve has opened.

From Fig. 8.12 it is clear that the higher the controlling pressure, the lower the variance (compare Δt_1 when p'_{ynp} and Δt_2 when p''_{ynp}).

The value of the controlling pressure at the moment of the start of valve opening is determined, apart from the spring force, by the value of the propellant pressure at the valve inlet.

Figure 8.13 shows curves of the change in time (with consideration of possible variances) of forces, acting on the moving system during valve opening (during the release of the controlling pressure). Here A and Б are curves of the drop in the resulting forces from the controlling pressure p_{ynp}; I-IV are curves of the rise in the resulting forces from the propellant pressure and the spring force. At the moments of intersection of curves A and Б with curves I-IV valve opening begins. The higher the propellant pressure, the higher the value p_{ynp} at which valve opening takes place and the lower the stability. When p_{ynp}= E (see Fig. 8.13) the variance is very small.

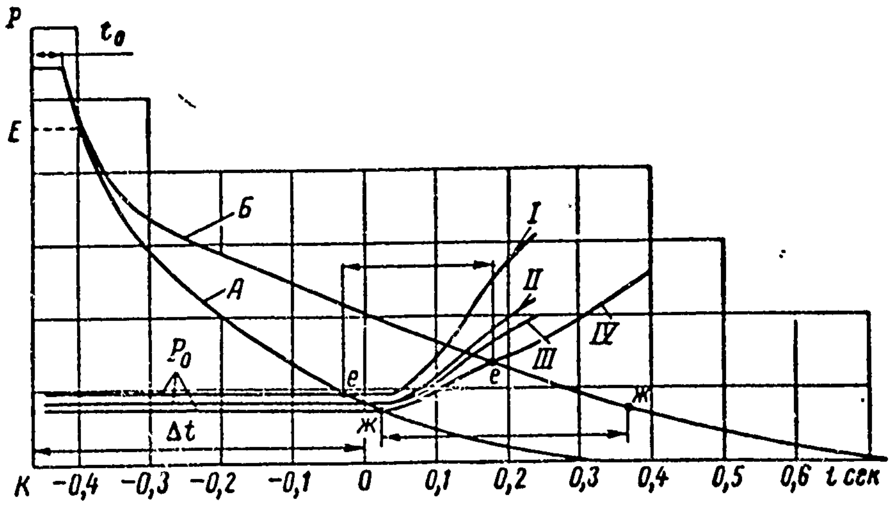

Fig. 8.13. Effect of valve holding time under oxygen, of initial pressure and of the rate of rise in the oxygen pressure on the moment of opening of an oxygen valve: I, II, III, and IV growth curves of the resulting forces from the oxygen pressure (depending on the acceleration regime of the TPA) and from the spring force; A, Б - curves of the fall of the resulting forces from the controlling pressure (depending on the valve's holding time under oxygen); P_0 - value of the force from the initial pressure of the oxygen in front of the valve; K - command to open the valve; 0 - the start of turbine operation.

In the given case the effect of an increase in the propellant pressure as a result of the start of operation of the TPA, in spite of the possible fluctuations in the law of the growth of this pressure, is more favorable for valve stability than the presence of a constant but low value of the propellant pressure (see segment e-e and ж-ж in Fig. 8.13). Of great significance is the constancy of time Δt between the start of operation of the TPA (point 0) and the command to open the valve (point K).

The process of opening of a normally closed valve, designed for low-boiling propellant, occurs differently than the opening of a normally open valve. When the controlling pressure is supplied, the air does not manage to be cooled, and therefore the

stability of such a valve is similar to the stability of a valve designed for high-boiling liquids. This also refers to the processes of closing of normally open valves. During the closing of normally closed valves (when the air pressure is released from the controlling cavity) the factors enumerated in this section take place to one degree or another – the degree of their influence on the stability depends on the time in which the valve is in the open state, i.e., on the degree of cooling of the air in the controlling cavity.

The most radical measure combatting instability of valves for low boiling liquids is the replacement of the controlling air by another gas, which is liquefied at a substantially lower temperature than air, for example, by helium.

The liquefaction temperature of helium at a pressure equal to 1 at is -268.9°C (let us remember that air at this pressure is liquefied at a temperature around -194°C). Thus, all problems connected with the liquefaction of a gas in a liquid cavity disappear. The valve's opening time will not depend on the time that it is kept under oxygen.

There is still another characteristic peculiarity of helium, under identical conditions (at a temperature above 0°C) the time of discharge of the helium from the controlling cavity will be significantly less than the discharge time of air or nitrogen. This is explained by the large value of the gas constant of helium [R of helium is equal to 212 (kgf·m)/(kg·deg); R of air equals 29.3 (kgf·m)/(kg·deg)] and, moreover, by the greater adiabatic factor (in helium, as in a monatomic gas, k = 1.66; in air k = 1.41). As a result of this the critical escape rate in helium is higher than in air [in helium - 873 m/sec, in air - 313 m/sec (at a temperature of 20°C)]. However the viscosity of helium is approximately equal to that of air.

Experimental work has shown that the time of discharge from the controlling cavity of the valve where the air has been replaced by helium is shortened at a temperature of 15-25°C by 30-60% and when the valve is handling liquid oxygen - by 4-7 times.

Figure 8.14 shows oscillograms of the valve's opening after a 15-minute and two-hours stay under liquid oxygen while using helium as the controlling gas (the valve is the same, in which the oscillograms shown in Fig. 8.10 were obtained - without the flow of gas through the controlling cavity).

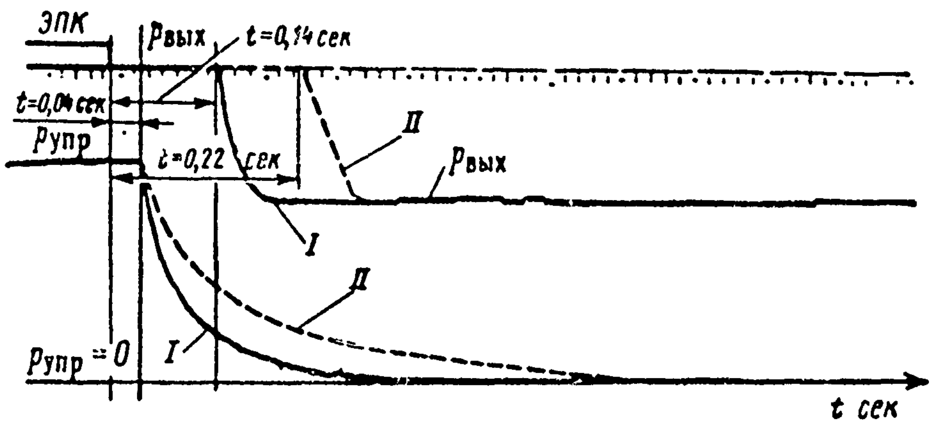

Fig. 8.14. Characteristics of opening of an oxygen valve (change of $p_{упр}$ and $p_{вых}$) using helium as the controlling gas: I - after holding the valve under liquid oxygen for 15 minutes; II - after holding the valve under liquid oxygen for two hours.

Under these conditions of handling helium the results shown in Table 8.4 were obtained (for comparison, the results of the work with air are shown).

It is somewhat more difficult to ensure the hermeticity of the joints when working with helium.

Table 8.4.

Time with holding the valve under oxygen, min	Time from the moment of giving the command until the moment of the start of valve opening, sec	
	with helium	with air
15	0.14	0.65
120	0.22	1.66
Working with water (instead of oxygen)	0.13	0.30

Of the other measures to combat instability (handling compressed air), apart from introducing a gas flow and increasing the controlling pressure at the moment of valve opening, we can cite the heating of the controlling cavity. Heating can be effected either by an air blast blown around the bellows, or with the aid of an electric preheater. The effectiveness of this method depends on the quantity of consumed heat energy.

Other methods of increasing the stability of pneumatic valves for low-boiling liquids are based on the above-described regularities, which are common to all pneumatic assemblies.

The most effecient method of increasing the stability is to reduce the volume of the controlling cavity and to sharply increase the cross-sectional communication lines, through which controlling air drainage takes place. Thus, after increasing the flow passage cross-sectional areas of the main line (of the tubing and of the EPV) from a diameter of 10 mm to a diameter of 32 mm, the deviation of the valve's opening times, the initial data for which are given in Fig. 8.11 and Table 8.3, came to a total of 0.05-0.06 sec

at p_{ynp} = 55 at. The opening time did not exceed 0.2 sec. However, this method leads to the encumbering of the engine with large-diameter tubes.

Another effective method is to reduce the controlling pressure (within the allowable limits, without adversely affecting the hermeticity of the valve). This brings about a reduction in the gas boiling temperature, and consequently, also a reduction in the quantity of liquefied air; the overall weight quantity of the gas is also reduced; the time of discharge and its variance will be reduced.

Stability of opening of an oxygen valve (see Fig. 2.9) for operation on main-stage mode is achieved by the independence of the explosive force of the bolt and of the spring force from the temperature and by the stability of the law of the pressure build up at the valve inlet.

The degree of balance of the valve exerts an influence on the stability of all the pneumatic assemblies. The use of a discharge system increases the value of the friction forces of the sealing elements with respect to the force from the propellant pressure, but reduces the value of the controlling pressure and reduces the volume of the controlling cavity.

Where normally closed valves are used, holding of the seals of the controlling cavity under pressure is eliminated, which eliminates the effect of a change in the density of the controlling gas on the valve's operating stability; however, it may be pointed out here that the moving system will exist in a state of prolonged rest, which will increase the friction force and its variance. Therefore, when using such valves (if this is possible with respect to the operating conditions) before filling with the component or starting the engine, the valve should be technologically operated.

In all cases, in order to ensure valve operating stability, it is desirable to use bellows in place of rings or seal rings.

APPENDIX

CALCULATING THE ESCAPE OF THE COMPONENT DURING PRESSURE DISCHARGE OF THE PRESSURIZED TANKS

In making devices for the neutralization of the drainage ejections it is necessary primarily to determine the gravimetric quantity of the product removed from the tank during drainage - with the release of the boost pressure, created by the air pressure.

Two initial states are possible:

1) no liquid product in the tank - it is completely consumed, i.e., there is only one phase in the tank - the gas mixture of the propellant and air vapors, existing at boost pressure p_H;

2) there are two phases in the tank at boost pressure p_H - apart from the gas mixture, consisting of component and air vapors, there is still the liquid component, capable of evaporating with a decrease in partial pressure of the component vapors.

Let us examine both cases.

1. Determination of the Quantity of Component, Drawn Off During Drainage of the Tank, in Which the Liquid Product is Absent

The quantity of vapors of the product Π_H (in kg), found in the tank at boost pressure p_H (in kgf/m^2), is expressed thus:

$$\Pi_{н} = \frac{p_{к}V}{R_{к}T_{н}},$$

where T_{H} is the initial temperature of the gas mixture in °K;

p_{H} is the partial pressure of the component vapors at temperature T_{H} in kgf/m^2;

R_{H} is the universal gas constant of the component vapors;

V is the volume of the gas mixture (volume of the tank).

The overall quantity of the gas mixture G_{H} (in kg) may be expressed thus:

$$G_{н} = \frac{p_{н}V}{R_{см}T_{н}},$$

where R_{CM} is the universal gas constant of the mixture.

With the draining of the tank to pressure p_{OCT} the quantity of mixture in the tank is reduced.

The remaining quantity of mixture G_{OCT} may be thus expressed:

$$G_{ост} = \frac{p_{ост}V}{R_{см}T_{ост}},$$

where T_{OCT} is the temperature of the mixture after drainage.

If we assume that the temperature in the tank differs greatly from the component condensation temperature, then the value of the gas constant of the mixture will not vary during drainage.

Thus, the quantity of the mixture in the tank is reduced by m times, where

$$m = \frac{G_{н}}{G_{ост}} = \frac{p_{н}T_{ост}}{p_{ост}T_{н}}.$$

The gravimetric quantity of component vapors will also be reduced by m times:

$$\Pi_{\text{ост}}^{\cdot}=\frac{p_{\text{к}}V}{R_{\text{к}}T_{\text{ост}}}\,\frac{p_{\text{ост}}}{p_{\text{н}}}.$$

Product vapors M (kg) will enter the neutralized apparatus in the quantity:

$$M=\Pi_{\text{н}}-\Pi_{\text{ост}}^{\cdot}=\frac{p_{\text{к}}V}{R_{\text{к}}T_{\text{н}}}\left(1-\frac{p_{\text{ост}}T_{\text{н}}}{p_{\text{н}}T_{\text{ост}}}\right).$$

In the general case with the outflow of the mixture from the tank the temperature of the mixture in the tank varies according to the polytropic curve p/γ_n = const.

In this case

$$T_{\text{ост}}^{\cdot}=T_{\text{н}}\left(\frac{p_{\text{ост}}}{p_{\text{н}}}\right)^{\frac{n-1}{n}};$$

then

$$M=\frac{p_{\text{к}}V}{R_{\text{к}}T_{\text{н}}}\left[1-\left(\frac{p_{\text{ост}}}{p_{\text{н}}}\right)^{\frac{1}{n}}\right]. \qquad (1)$$

Ordinarily drainage occurs slowly, and the temperature inside the tank remains almost unchanged as a result of the addition of heat from the environment, i.e., $T_{\text{ОСТ}} = T_{\text{Н}}$.

In this case n = 1 and then

$$M=\frac{p_{\text{к}}V}{R_{\text{к}}T_{\text{н}}}\,\frac{p_{\text{н}}-p_{\text{ост}}}{p_{\text{н}}};$$

if $p_{\text{ОСТ}} \ll p_{\text{Н}}$, then $(p_{\text{Н}}-p_{\text{ОСТ}}) \approx p_{\text{Н}}$.

Taking this into account, we obtain

$$\Delta M=\frac{p_{\text{к}}V}{R_{\text{к}}T_{\text{н}}}. \tag{2}$$

2. Determination of the Component Quantity, Removed During Drainage of the Tank, in Which There is a Liquid Product

Drainage of such a tank is usually carried out through the small discharge jets very slowly, since with an abrupt pressure discharge through the large flow passage cross sectional areas a great quantity of liquid component will be removed along with the gas.

Hence we will assume that the value of the partial pressure of the component vapors always corresponds to the temperature, i.e., in view of the fact that the discharge occurs slowly, the propellant manages to vaporize, and an equilibrium composition of the mixture is ensured in the gaseous phase:

$$p_{\text{к}}=f(T).$$

If this is so, then in draining the tank for time Δt there will be removed from the tank ΔG kg of the gas mixture, where the gas mixture will contain ΔM kg of the product:

$$\Delta M=k\Delta G,$$

where k is the concentration of the component in the mixture

$$k=\frac{p_{\text{к}}R_{\text{см}}}{p_{\text{см}}R_{\text{к}}}.$$

The quantity of gas mixture ΔG (in kg), removed from the tank during time Δt, can be determined as

$$-\Delta G = G - G',$$

where G is the quantity of mixture in the gas cushion at the given moment;

G' is the quantity of mixture in the gas cushion after the time interval Δt.

Then $$\Delta G = \frac{p_{\text{см}} V_{\text{п}}}{R_{\text{см}} T} - \frac{p'_{\text{см}} V_{\text{п}}}{R'_{\text{см}} T'} = \frac{V_{\text{п}}}{R_{\text{см}} T}\left(p_{\text{см}} - \frac{R_{\text{см}} T}{R'_{\text{см}} T'} p'_{\text{см}}\right),$$

where $V_{\text{п}}$ is the volume of the gas cushion (and not of the tank);

T', $p'_{\text{см}}$, $R'_{\text{см}}$ are the parameters of the gas after the lapse of time Δt.

Let us assume that $R'_{\text{см}} = R_{\text{см}} = \text{const}$; at high boost pressures $R_{\text{см}}$ changes negligibly little; for many liquids, in which the molecular weight is close to the molecular weight of air, the change in the universal gas constant of the mixture is also insignificant at low boost pressures. Since

$$T' = T\left(\frac{p'_{\text{см}}}{p_{\text{см}}}\right)^{\frac{n-1}{n}},$$

then

$$-\Delta G = \frac{V_{\text{п}}}{R_{\text{см}} T}\left[p_{\text{см}} - \left(\frac{p_{\text{см}}}{p'_{\text{см}}}\right)^{\frac{n-1}{n}} p'_{\text{см}}\right],$$

or

$$-\Delta G = \frac{V_{\text{п}} p_{\text{см}}}{R_{\text{см}} T}\left[1 - \left(\frac{p'_{\text{см}}}{p_{\text{см}}}\right)^{\frac{1}{n}}\right].$$

Since $p'_{см} = (p_{см} - \Delta p)$, then, excluding $p'_{см}$, we get

$$-\Delta G = \frac{V_п p_{см}}{R_{см} T}\left[1 - \left(1 - \frac{\Delta p}{p_{см}}\right)^{\frac{1}{n}}\right].$$

Expanding expression $\left(1 - \frac{\Delta p}{p_{см}}\right)^{\frac{1}{n}}$ into a binomial series, we get

$$\left(1 - \frac{\Delta p}{p_{см}}\right)^{\frac{1}{n}} = 1 - \frac{1}{n}\frac{\Delta p}{p_{см}} + \frac{\frac{1}{n}\left(\frac{1}{n} - 1\right)}{1 \cdot 2}\left(\frac{\Delta p}{p_{см}}\right)^2 + \ldots$$

and so forth.

Disregarding all the terms beginning with the third, as terms of a higher order of smallness relative to the value $\Delta p/p_{см}$, we get

$$-\Delta G = \frac{1}{n}\frac{V_п \Delta p}{R_{см} T}.$$

Hence

$$\Delta M = -\frac{1}{n}\frac{p_к V_п \Delta p}{R_к p_{см} T},$$

or

$$dM = -\frac{1}{n}\frac{p_к V_п}{R_к T}\frac{d p_{см}}{p_{см}}.$$

During time t the pressure in the gas cushion falls from $p_н$ to $p_{ост}$ and therefore

$$M = -\int_{p_{см} = p_н}^{p_{ост}} \frac{1}{n}\frac{p_к V_п}{R_к T}\frac{d p_{см}}{p_{см}}. \qquad (3)$$

It is not possible to take the integral in this form, since in the general form the value of $p_н = f(p_{см})$ or $p_н = f(T)$ is unknown.

For the particular case where the temperature in the tank (as a result of the import of heat to the tank walls) does not vary (the actual processes are close to this), we get an isothermic process.

For certain propellants the value of $p_н$ at the possible operating temperature (on the order of 50°C) is higher than atmospheric pressure. In these cases the pressure $p_{ост}$ is higher than the pressure of the environment.

With the use of low-boiling propellants (for example, liquid oxygen), in which at ordinary temperature the value $p_н$ is very great, measures are taken for thermal insulation of the walls of the tank from the environment, ensuring a low temperature of the product. In the majority of cases the removal (drainage) of the propellant vapors occurs continuously.

With an isothermic process

$$T = T_н = \text{const} \text{ and } p_к = f(T) = \text{const}; \quad n = 1.$$

Then it is easy to take the integral, and we obtain

$$M = \frac{p_к V_п}{R_к T_н}(\ln p_н - \ln p_{ост}). \qquad (4)$$

Assuming $p_{ост} = 1$ at[1], we obtain

[1]Naturally in the case where $p_н < 1$ at; otherwise, $p_{ост} > p_н$.

$$M = \frac{p_{\kappa} V_{\pi}}{R_{\kappa} T_{\text{н}}} \ln p_{\text{н}}, \tag{5}$$

where p_{κ} is in kgf/m^2 and $p_{\text{н}}$ is in kgf/cm^2.

A change in (increase) $R_{\text{см}}$ in proportion to the pressure drop, which we disregarded in the derivation of the formula, only increases the value of M.

However, in practice during drainage (even slow drainage) the temperature is reduced and (this is the main thing!) the actual partial pressure of the component vapors is lower than the value, corresponding to the pressure of the vapors at the actual temperatures, since vaporization of the propellant does not occur completely.

The practical value of the product removal will be less than that calculated from formulas (4) or (5).

Therefore, a coefficient of correction α is introduced:

$$M = \alpha \frac{p_{\kappa} V_{\pi}}{R_{\kappa} T_{\text{н}}} \ln p. \tag{6}$$

The physical concept of coefficient α may be represented from expression $\alpha p_{\kappa}/T_{\text{н}}$ - this is the actual average (during the process of draining) pressure of the vapors of the component, relative to the initial temperature of the gas in the tank. Value $\alpha < 1$. Maximum product removal will occur at $\alpha = 1$.

Formula (2) for the drainage of the tank without liquid can be represented in the form of formula (6); in this case the coefficient α will be expressed as 1/ln p.

Thus, formula (6) can be used when $p_к < p_{атм}$ for all cases of an isothermic process, where the value of coefficient α can vary from 1/ln p to 1.

When there are two faces in the tank, the coefficient α will depend on: the properties of the coefficient; the rate of drainage; the value $\Delta T = T_к - T$, where $T_к$ is the temperature of boiling of the component at atmospheric pressures; on the shape of the tank and level of the component; on the thermal resistance of the walls of the tank and so forth.

For the product freon-11? with a spherical tank, small amount of liquid, when t = 20-40°C and with excess pressure $p_н = 10$ at the value $\alpha \approx 0.4$.

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